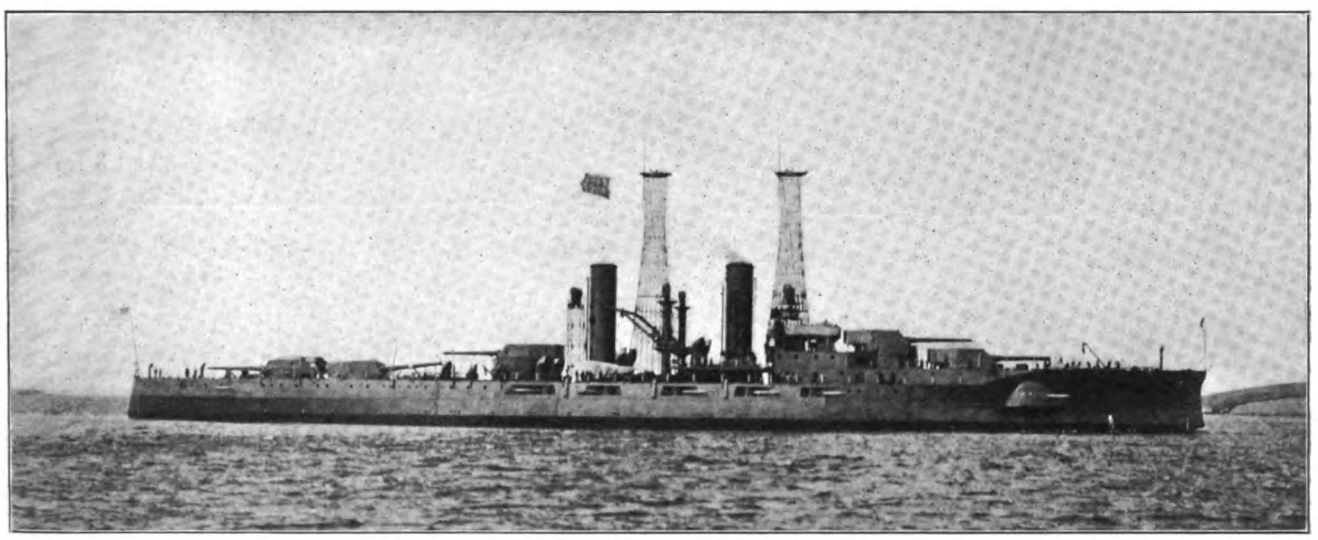
International Marine Engineering

DECEMBER, 1909.

UNITED STATES BATTLESHIPS DELAWARE AND NORTH DAKOTA.

The trials of the two 20,000-ton American Dreadnoughts *Delaware* and *North Dakota* were carried out during the latter part of October and the first part of November over the Government course at Rockland. These two battleships were authorized in 1906. The *Delaware* was built by the Newport News Shipbuilding & Dry Dock Company, Newport News, Va., and the *North Dakota* by the Fore River Shipbuilding Company, Quincy, Mass. The two ships are sister ships, and are exact counterparts as regards their principal dimensions and main features, but the *Delaware* is propelled by twin screws, driven by triple-expansion reciprocating engines, and the *North Dakota* by twin screws, driven by Curtis turbines. Each ship is 510 feet long on the waterline and 518 feet 9 inches long over all. The beam is 85 feet 2⅝ inches and the mean draft 27 feet. The normal displacement is 20,000 tons, and the full-load displacement 22,075 tons.



U. S. S. NORTH DAKOTA LEAVING BOSTON HARBOR FOR HER TRIALS OVER THE ROCKLAND COURSE.

In outward appearance the modern tendency in warship design is at once apparent in these ships. The Dreadnought idea of armament, with all big guns of a single caliber, is fully carried out, there being ten 12-inch 45-caliber guns mounted in pairs in revolving turrets on the center line of the ship. Much of the superstructure which was formerly visible in battleships is absent, and only a minimum amount of exposed area is visible with the exception of the guns themselves and their turrets. The appearance of the ship is further modernized by the use of the skeleton framework, fire-control masts, now standard in the United States navy, separate elevated searchlight platforms and the absence of any bridge decks except the single conning tower and navigating bridge.

As can be seen in the photograph, the vessels have a high forecastle deck and a wide flare above the waterline at the bow, qualities which should make for seaworthiness in heavy weather. The freeboard forward is 27 feet. The guns in the two forward turrets are, therefore, brought to an exceptional height above the waterline, the forward pair being 33 feet and the pair immediately aft 41 feet above the normal waterline. One of the after turrets also has an elevation of 33 feet above the waterline, enabling its guns to be fired over the two aftermost turrets, which are at such a height that their guns are 25 feet above the normal waterline. This arrangement of the main battery is one which is being copied to a certain extent by other navies, and which has much to recommend itself. All the guns have an exceptionally wide arc of fire, and all can be fired on either broadside. The fore-and-aft fire is limited to four 12-inch guns, whereas most of the English Dreadnoughts are able to bring six guns to bear directly ahead or directly astern. A very small angle of deviation from right ahead or right astern, however, is sufficient to bring this number down to four.

In addition to the main battery of big guns there are fourteen 5-inch rapid-fire guns for anti-torpedo boat attack, four 3-pounders, four 1-pounder semi-automatics, two 30-caliber machine guns, and two 3-inch field guns. There are also two 21-inch submerged torpedo tubes.

These battleships are among the most completely armored in the world. The main belt, which is 7 feet 6 inches wide, is 10 inches thick at its bottom edge and 12 inches at its upper edge. Above this, amidships, is a second belt, 8 feet wide and 10 inches thick at its bottom edge and 8 inches thick at the top. The main turrets and barbettes are protected by 11-inch armor, and the battery of 5-inch guns, which is mounted on the gundeck, by a belt of 5-inch armor.

Both ships were designed for a speed of 21 knots, the designed horsepower being 25,000. Twelve Babcock & Wilcox watertube boilers are used in each case. The normal coal supply is 1,016 tons, and the maximum 2,340. Oil fuel is also provided, to be used in conjunction with the coal.

One of the most important points about these vessels is the opportunity afforded for an exact comparison of the two types of propelling machinery, which is made possible by the simi-

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larity of the two ships. Details of the engines of the *Delaware* are not available, and only very general reports of her official trials. We are able, however, to give a somewhat more extended description of the turbine engines of the *North Dakota,* and a fairly detailed account of her trial trip.

A very complete description of the *North Dakota's* turbines was published on page 184 of our May, 1909, issue, but a brief recapitulation of the main points may not be out of place here. The turbines were designed to develop 12,500 horsepower, each at 245 revolutions per minute, with a steam pressure of 265 pounds per square inch in the steam chest and 28 inches vacuum in the exhaust shell. Each turbine is 144 inches pitch diameter, and 22 feet 6 inches long center to center of the main bearings. The expansion of the steam is divided into nine stages in the ahead turbines and two stages in the reverse turbines, the ahead and the reverse turbines being incorporated in the same casing.

In the first expansion stage of the turbines there are four rows of moving buckets on the wheel, since the greater energy drop in this stage produces a greater velocity of the steam jet from the nozzles, which requires more rows of buckets to properly absorb the energy at the bucket speed used. One-fourth of the available energy of the steam is expended in the first stage and three-thirty-seconds in each of the other stages. This is done in order to keep the pressure in the shell as low as possible. It requires, however, that the first-stage nozzle shall be of the expanding type, while all the other nozzles are of the parallel-flow type. The moving buckets for the sixth, seventh, eighth and ninth stages are all mounted on a single drum, there being three rows of buckets to each stage.

The reverse wheels are mounted in the after end of the casing, and, under ordinary conditions, when the turbine is running ahead, they are in a vacuum, and therefore do not waste power by steam friction. Cast steel steam chests for ahead and astern running are attached to the front and back casing heads, and are flanged for main steam pipes 13½ inches in diameter. The exhaust is through a rectangular opening, 4 feet by 10 feet, in the top of the casing at one side of the center line.

A regular marine thrust bearing is attached to the forward end of the turbine shaft, the thrust block forming an extension of the forward main bearing. In addition to taking any unbalanced thrust which may occur, this bearing also maintains the proper axial position of the rotor, so that the axial clearance of the blades is correct. This clearance is about 1/10 inch on the first wheel and increases as the size of the blades increases. The thrust bearing is placed at the forward end, so that any unequal expansion of the shaft and casing will be allowed for at the after end, where the clearance is largest.

The steam pressure at the forward end of the drum approximately balances the thrust of the propeller, so that the thrust bearing is only required to take the resulting unbalanced thrust, which is comparatively small.

The principal results from the official trials of the *North Dakota* were as shown in the tables.

OFFICIAL TRIALS OF THE U. S. S. NORTH DAKOTA.

	**8 Hours of Full Power Trial.	24-Hour Trial at 19 Knots.	24-Hour Trial at 12 Knots.
Actual average speed	21.64	*19.24	12.05
Revolutions per minute	280.4	231.9	143.2
Shaft horsepower of main turbines	31,400	*16,710	3,800
Indicated horsepower of engineer's auxiliaries	1,100	*660	400
Water rate of main turbines only	13.6	14.11	20.5
Water rate for all engineer's purposes based on total horsepower	13.96	15.29	22.3
Coal used, pounds per hour	54,400	*27,550	9,820
Coal used, tons per 24 hours	583	295.3	105
Coal per hour per shaft horsepower of turbines	1.74	*1.65	2.56
Coal per hour per horsepower for total horsepower	1.68	*1.58	2.34
Coal per hour of equivalent indicated horsepower based on 8 percent friction for reciprocating engine	1.55	*1.46	2.15

* All preceding figures are official except those starred.
** At the end of three hours a boiler tube burst, necessitating the cutting out of four boilers.

STANDARDIZATION TRIAL.

	Knots.
Highest speed on mile, uncorrected for tide	22.25
Mean of five high runs	21.83
Revolutions per minute for 21 knots	263.
Revolutions per minute for 19 knots	228.8
Revolutions per minute for 12 knots	142.5
Maximum shaft horsepower developed on mile	35,150.

POUNDS OF WATER PER HOUR PER HORSEPOWER FOR THE CURTIS TURBINE SPECIFIED AUXILIARIES.

	Specified.	Actual.	Saving, Percent.
3 hours of full power trial	15.1	13.96	7.5
24-hour 19-knot trial	16.1	15.29	5.0
24-hour 12-knot trial	23.2	22.3	3.9

Aside from her attainment of a full power speed of 21.64 knots, as compared with 21 knots specified, the most noticeable feature of these trials is the remarkably low coal consumption, particularly at slow speed, and the steaming radius made possible thereby. With maximum displacement this figures 9,000 nautical miles at 12 knots speed, while it is 4,600 nautical miles at 19 knots speed, and 3,000 nautical miles at a maximum of 21½ knots speed.

Compared with her sister ship, the *Delaware,* equipped with reciprocating engines, the *North Dakota,* with Curtis turbines, required only 295 as against 315 tons of coal per twenty-four hours on the 19-knot trial, and 105, as against 111 tons, on the 12-knot trial.

The trials of the *Delaware* were carried out just previous to those of the *North Dakota,* and with the following results:

On the four-hour full-speed trial, steaming with forced draft, the engines developed 28,600 indicated horsepower, driving the ship at an average speed of 21.56 knots. The maximum speed for 1 mile was 21.98 knots. During the four-hour trial, unofficial figures indicate that the average revolutions of the engines were 127, the coal consumption per indicated horsepower working out at 1.8 pounds, and the water consumption of the main engines at 13.5 pounds. The mean speed for five separate runs over the measured mile was 21.44 knots, with a corresponding horsepower of 28,578.

Of course, in comparing the horsepower of the reciprocating engines and turbines the difference between brake-horsepower and indicated horsepower must be kept in mind, the former being in the neighborhood of 88 percent of the latter. One fact is particularly emphasized in the performance of these two vessels, and that is the possibility of forcing turbine engines far beyond their designed power. In a turbine-driven ship every ounce of steam that can be generated can be applied to propulsion, vastly increasing the power of the turbines, and, consequently, the speed of the vessel; whereas, on a ship driven by reciprocating engines, it is not uncommon for the boilers to generate more steam than can be used by the engines, and hence it is impossible to get as good results from forcing.

Due to the lack of official figures, it is impossible to compare the steaming radii of the two vessels at various speeds. It is claimed, however, that at 12 knots the *Delaware* has a steaming radius of 9,300 nautical miles, whereas the *North Dakota* has a radius of only 9,000 nautical miles. Similarly, at 19 knots, the figures are 5,400 for the *Delaware,* 4,000 for the *North Dakota,* and at a speed of 21 1/7 knots 3,170 for the *Delaware* and 3,000 for the *North Dakota.*

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THE DESIGN OF REVERSING ENGINES.

BY EDWARD M. BRAGG, S. B.

Reversing engines are required upon marine engines of any size. While the smaller engines may be reversed by hand, the attempt to do this in engines over 1,000 indicated horsepower would necessitate such a large multiplication of power that the operation would be too slow for safety.

Some of the types of reversing engines employed are shown in Figs. 1 to 4. There are two general types, the direct acting, shown in Figs. 1, 2 and 3, and the all round, shown in Fig. 4.

In Fig. 1, *A* is the steam cylinder, *B* is the oil-brake cylinder, *C* is the valve chest containing the valve which governs the admission of steam to *A*, and *D* is the valve chest which contains the valve governing the flow of oil from one side of the piston to the other. The oil cylinder is used to prevent the gear from getting too much momentum, and so doing damage when it is necessary to reverse quickly. It also causes the gear to move smoothly. Fig. 5 shows a section through the valve chest *C* of Fig. 1. In this case a piston valve is used for both steam and oil. The steam is taken in

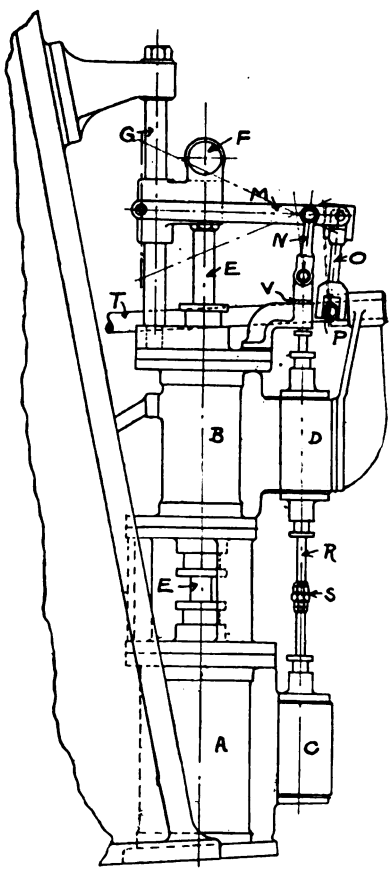

FIG. 1.

the middle of the valve and it exhausts at the ends. More commonly a slide valve is used, in which case the steam is taken at the ends of the valve and exhausts in the middle. The piston valve is so made that it overlaps the edges of each port by 1/16 or 3/32 inch. When no oil cylinder is used, as in the gear shown in Fig. 2, the exhaust lap is usually made larger, so that considerable compression may be obtained at the end of the stroke and produce the cushioning that the oil would. In the case of the valve for the oil cylinder the laps must be no more than are necessary for oil tightness, as the gear will be locked as soon as the valve closes.

In some cases the steam valve is made "line and line," or with no lap at all, and occasionally with a clearance. This is done when it is thought necessary to have the steam acting upon the piston in the "ahead" position to keep the links from working over towards mid position. Referring to Fig. 5, it will be seen that if we attempt to make a negative steam lap on the valve there shown by lengthening or shortening the valve stem with the adjusting nuts *S*, we shall increase the steam lap on the other end of the valve and throttle the steam when going from the "head" position to the "astern" position. In fact, it would be possible by this means to put so much clearance on the "ahead" position that the valve would not open the port to the other end of the cylinder.

The displacement of the valve is usually much less than the width of port. In Fig. 5 the lap of the valve is 3/32 inch, the width of the port is 7/8 inch, and the clearance is 1/4 inch, so

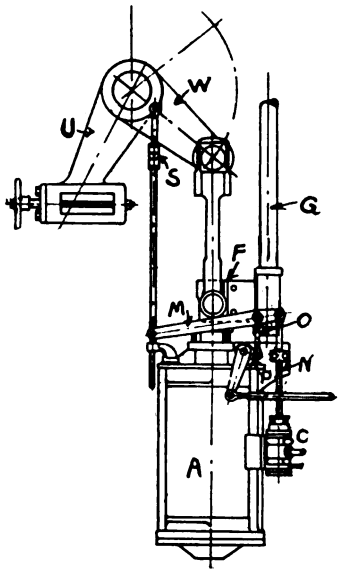

FIG. 2.

the maximum port opening of the valve is 5/32 inch. If the valve stem length were altered at *S*, Fig. 1, so that the lap on one end was increased from 1/32 to 1/4 inch, the valve would not open at that end. This maximum port opening is made small so that the links will stop moving almost as soon as does the reverse lever. The greater the maximum port opening the longer the links will keep moving after the reverse lever stops. The first movement of the reverse lever takes up the clearance at *V* in Fig. 1, and then it cannot move further until the crosshead commences to move. By following up the crosshead with the reverse lever the valve is kept open, and then when the lever is stopped a very slight movement of the crosshead brings the valve back to its mid position. The link *O* is placed where it is most convenient, and the lengths of the bell-crank levers made such that the movement of *O* is to the movement of the crosshead as the distance of *O* from the floating lever *N* is to the distance of the crosshead from that lever.

In the Brown gear shown in Fig. 3 the pitch of the coarse thread corresponds to the longer arm of the lever *M* in the other gears, and the pitch of the fine thread corresponds to the shorter arm. The crosshead *N* corresponds to the floating lever *N* in the other gears. This type usually has an oil cylinder above the crosshead *F*, and the upper end of the valve spindle *M* is attached to the valve of the oil cylinder, and regulates the flow of oil from one side of the piston to the other. The diameter of the oil cylinder is usually made about two-thirds of the diameter of the steam cylinder.

It will be found usually that the diameter of the reversing engine cylinder is from .2 to .185 the diameter of the low-pressure cylinder of the main engine. It will vary with the

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boiler pressure, steam speeds and eccentricity. The reversing engine takes its steam from the main steam line between the boiler and the throttle valve, so as to have steam even when the main engine is shut down. The size of the piston should be such that with .9 the boiler pressure there will be a twisting moment exerted equal to twice that for which the reverse shaft was designed. If the steam speeds are low the piston valves will be larger in diameter and the slide valves will be broader, so the frictional work to be overcome in moving the valves will be greater. The greater the eccentricity the greater the distance through which the valves must be moved. Some consideration should therefore be given to these three conditions in settling upon the size of the reverse cylinder. The stroke of the reversing engine is usually about 2.5 the travel of the valves, but varies from two to three times this amount.

The reversing engine lever, or levers (see *W*, Fig. 2), in going from the ahead to the astern position, must move through the same angle as do the reversing levers *V*, and the length of the former should be such that the angle be-

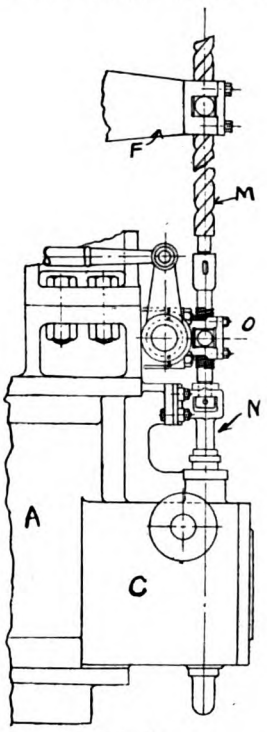

FIG. 3.

tween the extreme positions is subtended by a line whose length is equal to the stroke of the reversing engine. If Θ is this latter angle, and s is the stroke of the reversing engine, the length of the reversing engine lever will be:

$$l = \frac{s}{2 \sin \Theta} \qquad (1)$$

The piston rod can be calculated by the formula given for piston rods in the chapter on "Marine Engine Design." The formula is as follows:

$$D^2 = \sqrt{\frac{0.48\ F\ C\ l^2}{10{,}000{,}000}} + F^2 + F \qquad (2)$$

D = diameter of rod in parallel part.

$F = \dfrac{2\ W}{\pi\ f}$

W = load on rod.

f = allowable stress per square inch = C/N.

C = ultimate strength of material = 60,000 — 75,000 pounds.

N = factor of safety — 15.

l = length of parallel body of rod in inches = stroke + diameter of cylinder.

The connecting rod can be calculated by the formula[2] given in the same article for connecting rods:

$$D^2 = \sqrt{\frac{1.08\ F\ C\ l^2}{10{,}000{,}000}} + F^2 + F \qquad (3)$$

All the quantities are as above except that $N = 12$, and the length of the rod is usually from 1.5 to 2.5 the length of the reversing engine lever. This length must suit the position of

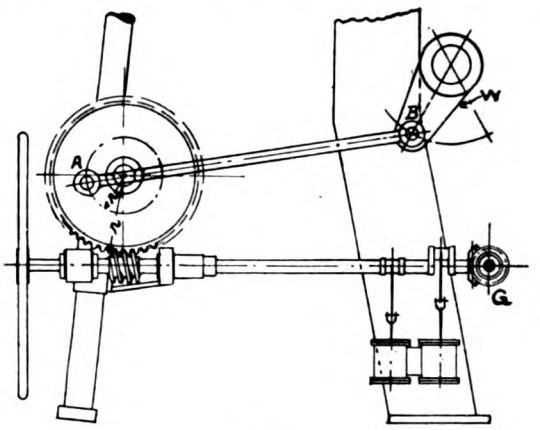

FIG. 4.

the cylinder relative to the reverse shaft. When an oil cylinder is used and the crosshead is above the oil cylinder the rod will have the lower value given; when the crosshead is between the two cylinders, the length may exceed that given above. The maximum force transmitted through the connecting rod will be slightly greater than that transmitted through the piston rod, due to the angle which the former makes with the center line of the engine at the middle and ends of the stroke. This increase is so slight, however, that

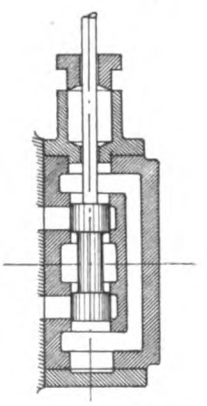

FIG. 5.

it can be neglected and the rod calculated for the same load as the piston rod. The diameter of the rod at the ends can be reduced to .85 of the diameter at the middle.

The pressure upon the crosshead pin can be as high as 3,000 pounds per square inch, since the connecting rod moves through a very slight angle, and the engine will not be used with sufficient frequency to heat the pin.

The crosshead guide is usually a circular rod, and should be designed to resist the bending that will come upon it when the piston is at the middle of its stroke. The center line of the engine is so placed that the connecting rod will swing

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through equal angles on either side. The maximum load coming upon the guide will be:

$$w = \frac{r - \sqrt{r^2 - (s/2)^2}}{2 \times l} \times W \qquad (4)$$

r = length of the reversing engine lever.
s = stroke of reversing engine.
l = length of connecting rod.
W = load upon piston rod.

The pressure per square inch on the crosshead guide should be from 60 to 80 pounds.

In the All-round gear shown in Fig. 4, the radius r should be about 2.5 times the eccentricity of the valves, and the radius r of the pitch circle of the worm should be such that with the pitch used the number of teeth on the worm wheel will be between 30 and 45. A less number than 30 gives weak teeth and more than 45 makes the time of reversing rather long. With 30 to 45 teeth the reversing engine will have to make from 15 to 23 revolutions to throw the links over if a single-threaded worm is used. The design of worms and worm wheels has been taken up under the head of Turning Engines, and some of the formulæ there given for the small worm and worm wheel can be used here.

Referring to Fig. 4 it will be seen that

$$\frac{T}{l} = \frac{mep \times a \times 2s \times r \times e \times e_1}{p\ r_1} \qquad (5)$$

and if $a = \frac{\pi d^2}{4}$, and $p = \frac{2 \pi r}{n}$,

$$d^2 s = \frac{4\ T\ r_1}{l \times mep \times n \times e \times e_1} \qquad (6)$$

T = twisting moment in inch pounds on reverse shaft necessary to throw all the gears.
l = length in inches of reverse engine lever.
mep = mean effective pressure in cyliders of gear.
a = area of cylinder in square inches.
s = stroke of engine in inches.
r = radius of pitch circle of worm wheel in inches.
r_1 = eccentricity of pin A in inches. (See Fig. 4.)
p = pitch of teeth in inches.
n = number of teeth on worm wheel.
e = efficiency of worm.
e_1 = efficiency of engine.

Formula (15), page 426, gives the stress at the root of the teeth of the worm wheel:

$$f = \frac{4.2\ F}{C_1\ p^2}$$

Formula (8) in the same issue gives:

$$F = .96\ Z\ \frac{1}{p};$$

and from (7) in the same issue:

$$Z = mep \times a \times s.$$

Therefore, $$f = \frac{4.03\ mep \times a \times s}{C_1\ p^2} \qquad (7)$$

From formula (5):

$$mep \times a \times s = \frac{T\ p\ r_1}{2\ l\ r\ e\ e_1}$$

Therefore, $$f = \frac{2.02\ T\ r_1}{C_1\ p^2\ l\ r\ e\ e_1} \qquad (8)$$

There seems to be no reason why f should vary inversely as the efficiencies of the gear and of the engine. Referring to formula (6), page 426, it will be seen that F is a function of these efficiencies. The presence of e and e_1 in the denominator of (8) will therefore cause the expression to be independent of these quantities. Assuming that $e = .4$ and $e_1 = .8$, we have:

$$f = \frac{6.3\ T\ r_1}{C_1\ p^2\ l\ r} \qquad (9)$$

Therefore, $$p = \sqrt{\frac{6.3\ T\ r_1}{f\ C_1\ l\ r}} \qquad (10)$$

p, T, r_1 r and l are as above.
f = allowable stress at root of teeth.
= 3,500-4,000 for cast iron and 5,000 for cast steel.
C_1 = breadth of wheel at root circle in terms of the pitch. C_1 is usually about 2.

(To be Concluded.)

APPLICATIONS OF ELECTRICITY TO PROPULSION OF NAVAL VESSELS.*

BY W. L. R. EMMET.

The figures and statements given in this paper are not mere theories based upon supposed possibilities, but in all essentials are accomplished facts, the nature of the case being such that we need not go beyond the scope of our actual experience to accomplish the purposes here proposed. These plans are actual designs worked out in every significant detail, which might be, and, if opportunity offered, would be contracted for and fully guaranteed.

The data and information given relate to two distinct methods, both adapted to the propulsion of battleships of the design of the *Arkansas* and *Wyoming*. Either plan is fairly indicative of possibilities in other war vessels, although each case must be designed on its merits if the best results are to be expected. In the first of these methods, which will be spoken of as "Combination Drive," generating units with motors and low-pressure turbines on propeller shafts are used together and separately for different conditions. In the second method, called here "Electric Drive," the propulsion is wholly by electric motors. The first of these plans was made the subject of a proposition to the Government for one of the new battleships, the turbine part being designed by the Fore River Shipbuilding Company and the electrical part by the General Electric Company. The second has been designed since the bids for these ships were opened.

In both of these designs the steam conditions specified by the Navy Department, namely, 260 pounds gage and 50 degrees F. superheat, have been used in calculating results. In both cases vacuum diminishing with load from 28.5 inches at 12 knots to 27 inches at 20.5 knots has been assumed. The Parsons curves given have been taken from guarantees without knowledge of vacuum proposed. If the vacuum is better than 27 inches, the Parsons water rates at the high speeds should be higher than those given by curves.

COMBINATION DRIVE.

With this drive it is proposed to use twin screws and to install upon each propeller shaft a low-pressure turbine and an electric motor, both of these being installed in the engine room provided for in the design of the ship. We would also install in each engine room a high-speed steam turbine generating set. The capacity of the generators and motors would be such that they would be capable of delivering, at 20.5 knots,

* Read before the Society of Naval Architects and Marine Engineers, New York, November, 1909.

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TABLE I.—COMBINATION DRIVE.

Knots	12	13	13	14	14	16	18	20	20.5	21
Shaft horsepower	4,400	5,500	5,500	6,900	6,900	10,450	15,350	22,700	26,000	30,700
Sum of motor output	4,400	5,500	5,500	6,900	6,900	8,750	10,000	11,250	11,700	12,350
Low-pressure turbine output						1,700	5,350	11,450	14,300	18,350
Number of poles on motor	42	42	42	42	28	28	28	28	28	28
Motor speed	131	142	142	153	153	175	198	224	232	240
Motor efficiency	94.5	95	95	95.5	95.5	95	95	95.5	95.5	95.5
Vacuum	28.5	28.5	28.5	28.5	28.25	28	27.75	27.25	27	27
Generator speed	1,400	1,510	1,510	1,625	1,085	1,250	1,370	1,580	1,645	1,710
Number of generators	1	1	2	2	2	2	2	2	2	2
Flow and leakage	52,300	62,700	67,000	82,500	92,000	133,700	195,700	276,300	315,000	366,500
Water rate for motor output	11.9	11.4	12.2	12.0	13.3	15.25	19.7	24.75	26.9	30.3
Low-Pressure Turbine										
Bucket efficiency						50	52	58	59.5	60
Flow						37,800	114,000	221,000	274,000	348,000
Water rate per shaft horsepower						22.2	21.4	19.30	19.1	19
Horsepower output						1,700	5,350	11,450	14,300	18,350
High-Pressure Part of Turbine.										
Bucket efficiency						61	64	67.5	69	70
Bucket water rate						30.5	29	27.5	27	26.6
High-pressure flow						39,700	119,500	232,000	288,000	365,500
Motor output						1,070	3,620	7,500	9,550	12,350
Motor output from cond. part						7,680	6,380	3,750	2,150	.....
Corresponding flow						93,000	75,200	44,300	26,000	.....
Total flow, including leakage	52,800	63,500	68,500	84,600	93,500	133,700	195,700	277,300	315,000	366,500
Water rate per shaft horsepower	12	11.6	12.5	12.2	13.5	12.8	12.75	12.25	12.1	12

two-fifths of the total power required for propulsion. The remaining three-fifths of the power under these conditions would be delivered by the low-pressure turbine, which at such a time would receive steam exhausted from the generating unit. This low-pressure turbine would also be fitted with two reversing stages, similar to those which would be adopted with direct-turbine drive. The low-pressure turbines would also be so arranged that they could take high-pressure steam from boilers through separate nozzles, and with such supply they would act as fairly efficient high-pressure turbines. With such high-pressure steam supply the ship would make a speed of about 19 knots with the same steam required for 20.5 knots with the combination drive.

This turbine has two exhaust openings, one connecting to the second stage and the other to the fifth stage. When the ship is running at about 20.5 knots all the steam would pass through the first of these exhaust openings to the low-pressure turbine, its pressure between the two being about 50 pounds absolute. When the ship operates at speeds below 15 knots the low-pressure turbine would not be used at all, the power being delivered to the propellers entirely by motor and being all generated by the high-speed turbine. At speeds between 15 and 20.5 knots, a part of the steam would pass to the condenser through the low-pressure turbine, and part through the lower

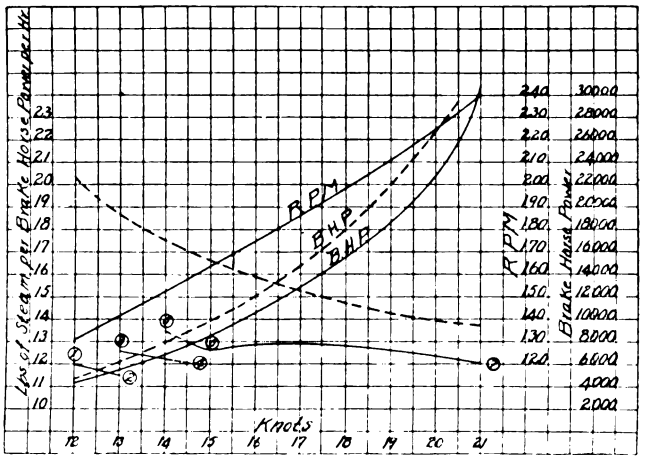

FIG. 1.—Curves showing steam consumption with combination drive compared with guarantees on Parsons equipment for the same ship. Also power curves for both cases and propeller speed curve for combination drive. Dotted lines, Parsons drive; full lines, combination drive; line 1-2, one generator and motors connected for 42 poles; line 3-4, one generator and motors connected for 28 poles; line 5-6, two generators and motors connected for 28 poles; line 6-7, combined effect of motors and low-pressure turbines. At 7, all steam passes through low-pressure turbine, at other points only a part.

stages of the high-speed turbine. The design of generating unit provides for a valve by which any desired number of third-stage nozzles can be closed, and by this means the division of steam between the low-pressure stages of the high-speed turbine and the low-pressure turbine can be controlled.

The curve sheet, Fig. 1, and the accompanying tabulation show all the relations of speed, power and efficiency.

It will be seen that in this plan the electrical apparatus acts simply as a speed-reducing bond between the generating turbines and the propeller shafts. The motors are not used for reversing, and are of the simple, squirrel-cage induction type. The voltage is low and the arrangement of generator and motor constitutes the simplest known means of electrical power transmission. With such apparatus, insulation trouble or mechanical trouble is practically unknown, and there is no other form of mechanism which can accomplish such results with equal simplicity and certainty.

Since in a warship it is desirable to operate efficiently at low speed as well as at high, the motors proposed in this case are so arranged that they can be connected either for 28 or 42 poles. In the higher-speed ranges the 28-pole connection is used, and in the lower-speed ranges we use the 42-pole connection. This change of connection is made by a single movement, which actuates a group of toggle switches, so designed that their action is perfectly positive and dependable. These switches could be so arranged that they could not be moved when the circuit was energized, so that no trouble could result from opening or closing them under load.

In this arrangement the electrical apparatus, with high-speed turbine, constitutes an auxiliary, the purpose of which is to improve efficiency under all speed conditions, and particularly to adapt the ship to economical operation at cruising speed. These results it will accomplish without the introduction of any feature which can be considered problematical. Turbine generating units exactly equivalent to that proposed are being widely used with better efficiencies than those here assumed, and motors of similar character, size and speed are delivering uniformly successful work under the most difficult conditions, such as mine hoisting, rolling mill work, etc., and mechanical or electrical trouble with either class of apparatus is of inappreciable extent.

This combination plan, as here explained, was decided upon largely with a view to overcoming the doubts and fears which might be raised by the proposition to introduce such a novelty into the new battleships. The arrangement might have been made lighter and less expensive by adapting the motors to reversal so that the reversing turbine could be left out. This would have necessitated changing poles in the ratio of 2:1 in-

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stead of as proposed, but the results obtainable with suitable designs would have been nearly the same. The reversal accomplished in this way would have been just as quick and effective as that afforded by the reversing wheels. Another feature of this scheme which it was hoped might allay the doubts and fears of uninformed persons is the fact that it could be readily convertible into a direct turbine drive.

ELECTRIC DRIVE.

Since the completion of these designs for combination drive and their rejection by the Government, the writer has designed an equipment somewhat similar in principle, in which the whole power is delivered to the propeller shafts by means of motors. On each propeller shaft it is proposed to install two motors, one of which is arranged with pole-changing switches, so as to adapt it to use at lower speeds, this motor being similar in character to that proposed with the combination drive. The other motor is adapted only to the smaller number of poles used with high speeds, and is arranged with a resistance connection to its motor so that it is suitable for producing the high torque desirable in quick changes of the ship's direction. The generating units proposed in this case are of a type representative of the highest development which has been attained. They are designed to give a very uniform efficiency through wide ranges of load and speed, so that they will accomplish the various functions desired with the best general effect. The voltage of these generators and motors is such as to give the simplest and most dependable windings for apparatus of the size, the maximum potential generated being about 2,200 volts.

The results in steam consumption at different speeds obtainable by this equipment are shown in Fig. 2. The water-rate curves there given refer to the propelling machinery alone, and do not include any other steam consumptions. The dotted curves on the same sheet show guarantees made on Parsons turbine equipment for the same ship. These Parsons curves presumably make some allowance over the results expected. It will be observed that the Parsons power curve is considerably higher than that assumed for the electric or combination drive. A large part of this difference is, however, certainly attributable to the fact that the proposed Parsons equipment operates with four propellers and much higher propeller speeds, which will give a lower efficiency to the propellers themselves and a lower propulsive efficiency of the ship, since one propeller is bound to interfere with the action of the other. The water-rate curve given for the Parsons equipment seems fairly reperesentative of the results claimed for the best Parsons designs. It agrees closely with the curve of water rates calculated for the Curtis turbine for a similar ship.

TABLE 2.—ELECTRIC DRIVE.

Knots	12	14	15	15	18	20	20.5
Shaft horsepower	4,400	6,900	8,600	8,600	15,350	22,700	26,000
Motors, number of poles	50	50	30	30	30	30	30
Motor speed	131	153	164	164	198	224	232
Motor efficiency	94.5	95	96	96	96	96	96
Motor power factor	60	65	83	79	84	85	85
Generator speed	1,105	1,295	830	830	1,005	1,130	1,172
Vacuum	28.5	28.25	27.50	28.25	27.75	27.25	27.0
Number of generators	1	1	1	2	2	2	2
Number of motors	2	2	2	4	4	4	4
Pounds of water per shaft horsepower	12.3	11.8	12.55	13.4	12.25	12.1	12.3

Propeller Speed.—In designing this electric drive we have adopted, for the sake of comparison, the same propeller speed used with the combination drive, and this in turn was taken for a similar reason from that proposed for direct propulsion by Curtis turbines. In the combination drive it will be desirable to keep to the best Curtis turbine speed, since a large proportion of the power is delivered by turbines. In the electric drive, however, we are not limited as to propeller speed,

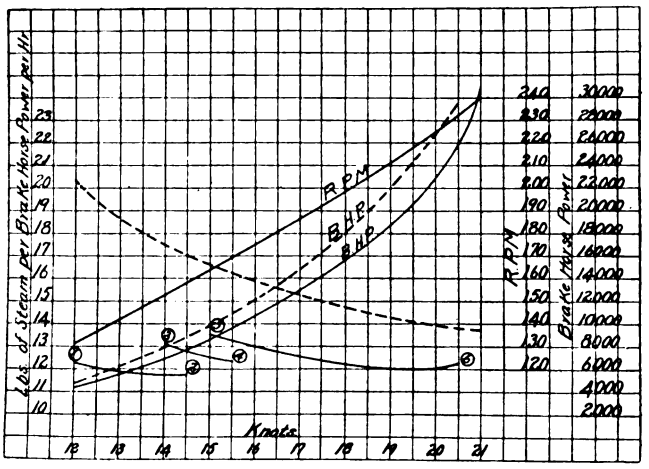

FIG. 2.—Curves showing steam consumption of electric drive compared with results guaranteed on Parsons equipment of the same ship; also power curves for both cases and propeller speed curve for electric drive. Dotted lines, Parsons drive; full lines, electric drive; line 1-2, one generator and two motors connected for 50 poles; line 3-4, one generator and two motors connected for 30 poles; line 5-6, two generators and four motors connected for 30 poles.

and it would be possible to adopt considerably lower propeller speeds, and by so doing to improve the net result without serious increase of weight.

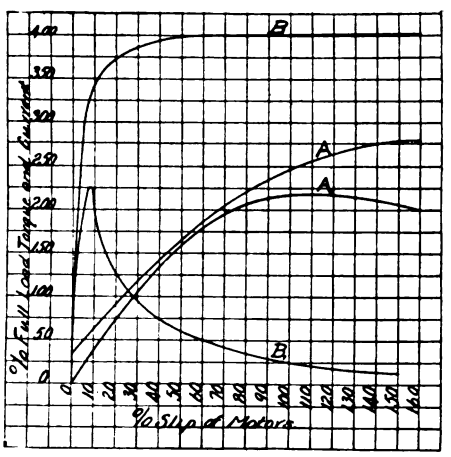

FIG. 3.—Speed, torque and current curves of motors with full frequency and voltage. *AA*—Current and torque with resistance. *BB*—Current and torque without resistance.

Data relating to the operation of this electric drive at different speeds are given in the accompanying tabulation.

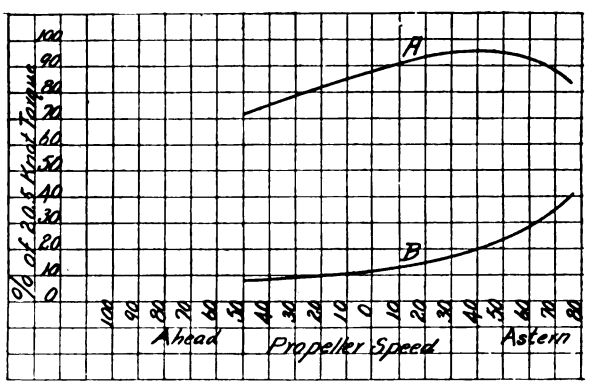

FIG. 4.—Torque and current curves of motor as used, with resistance inserted, in reversing ship. These curves assume normal frequency and reduced voltage. In practice, both frequency and voltage might be reduced, which condition would give better torque. Speed is given in percent of full frequency. Ahead means against generator; astern, with generator.

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Reversal.—The characteristics of the motors used in reversal of the propellers are shown by curves, Figs. 3 and 4. Fig. 3 shows the torque and current of such a motor under different speed conditions with full impressed voltage and frequency, two of the curves applying to the condition without resistance in the armature and the others applying to the condition with resistance in the armature. Fig. 4 shows the torque obtainable with such a motor with and without resistances when supplied with the normal full-load current of one generator at normal frequency. These motors would be capable of receiving for a short time even more current than this, but the torque here shown is far more than that provided by the ordinary reversing machinery for turbine ships. Curve *B* in Fig. 4 shows the relatively weak torque obtainable without resistance in the motor armatures. The results shown by this curve can be obtained from any of the motors; those on curve *A* can only be obtained from the motors which are arranged for the insertion of armature resistance. The armature resistances proposed would be connected to the motor through collector rings, and the resistance would be cut out by short circuiting the collector rings, so that no current would pass through brushes except when the resistance was in use. A form of resistance has been developed by experiment by which a very large power can be dissipated in a small space without complication or difficulty, and this with all other details has been included in weight estimates.

Weights.—The weight of the Parsons turbines alone proposed for these ships, as designed and estimated by the Navy Department, is given as 484.7 tons, which weight does not include any piping, bearings, shafting, valves or auxiliaries. The following figures show weights of equivalent parts with the combination drive and electric drive as here designed.

COMBINATION DRIVE.	Pounds.
Two generators	125,000
Two turbines	154,800
Two motors with pole-changing switches	140,000
Switches, etc.	2,000
Additional ventilating ducts	5,000
Two low-pressure turbines	740,000
Total	1,166,800
Equals	520 tons

ELECTRIC DRIVE.	Pounds.
Two motors with pole-changing switches	142,000
Two motors (M)	134,000
Switches, levers, supports, etc.	6,000
Cables, busses, supports, etc.	4,000
Rheostats	3,000
Two generators	280,000
Two turbines, without bearings	218,000
Additional ventilating ducts	7,000
Total	794,000
Equals	354 tons

From this it will be seen that, considering these parts alone, the combination drive would be 35 tons heavier than the Parsons equipment mentioned, and the electric drive would be 131 tons lighter. In the comparison of steaming distances which is here made, this difference alone is considered, but a consideration of the designs will show that other large economies in weight would be effected with the electric drive. The piping system necessary with the Parsons turbine equipment is very complicated, and the weight of steam and exhaust piping and valves given in the department's estimates amounts to 76.5 tons. With this electric drive only one steam pipe connection is necessary inside of the engine room. From this it will be apparent that there is a great saving of piping weight and complication in the electric drive as compared with the Parsons equipment. The writer has no means of knowing the necessary weight of piping outside of the engine room, but it is believed that saving in the engine room would amount to at least 40 tons.

The weight of boilers proposed for these ships is 555 tons, and if with the electric drive the boiler equipment was cut down in proportion to the saving in steam consumption at 20.5 knots, as indicated by the comparisons given above, a further saving in weight of about 108 tons would be effected. This estimate assumes that uses of steam outside of the prime movers are the same in both cases and in accordance with the department's estimates.

TABLE 3.

	Pounds Coal Available.	Water Rate.	Shaft Horsepower.	Distance in Knots at 12-Knot Speed.
Parsons	5,550,000	25.6	4,700	4,700
Electric	6,080,000	17.8	4,400	7,900

Cruising Distance.—The hull plans of these ships, as designed for Parsons turbines, show a bunker capacity of 2,476 tons, or 5,550,000 pounds. Adding coal in place of weight diminutions above mentioned, the total would become 6,080,000 pounds. If we assume 8.5 pounds of water evaporated per pound of coal, and the propeller, steam and electrical efficiencies above indicated, we have the comparison shown in Table 3.

If the same boiler equipment is retained for both systems the comparison will be as follows:

TABLE 4.

	Pounds Coal Available.	Distance at 12-Knot Speed.	Speeds with Equal Steam Flow.
Parsons	5,550,000	4,700	20.5
Electric	5,838,000	7,600	21.2

These figures assume that in both ships the steam consumption for all uses outside the main prime movers at 12 knots is 26,300 pounds per hour, this being the figure given in the Parsons guarantees.

It may be claimed that the Parsons turbine would do better than here assumed, the figures taken being guarantees which presumably afford some latitude for error. Comparison with other cases, however, does not indicate that the water rates assigned to this Parsons turbine are much too high. In the case of the *Lusitania*, where the high speed and large power afford ideal conditions for turbine drive, the water rate shown by tests at maximum speed was 12.77 pounds per shaft horsepower in the turbines alone with 28 inches vacuum, and with 27 inches vacuum it would presumably have been 13.5 pounds. A corresponding water rate shown by the Parsons curves here given, which form the basis of this comparison, is 13.75 pounds, with a horsepower output considerably less than half.

These figures show an increase of 69 percent in the cruising distance, and it is needless to comment upon the immense value of such a difference in any war vessel.

In this comparison it should be remembered that the performances of the electric equipment are very conservatively estimated, and that water rates and efficiencies on existing apparatus would justify the expectations of better performances. It should also be remembered that differences in weight of piping are not considered in this comparison, and also that there should be a large saving in weight of bearings and supporting structures if the electrical equipment is compared with the Parsons turbines. It should further be considered that the propeller speeds here adopted are not those most favorable to electric drive, they having been assumed for the sake of direct comparison with existing designs for the same ships. The electric drive would give almost the same efficiency with lower-speed motors, and the weight would be only slightly increased, while a very large improvement in propeller efficiency could be effected. To get the best results from such a new

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method, the whole ship should be designed to suit it, while in the designs here given it is simply made so that it could be substituted for the proposed combination drive without any change except in the propelling machinery itself.

Evidences.—The practicability of the plan of electric drive here proposed cannot be questioned, and the figures given must result from the expected efficiency in generating units and motors. The motor efficiencies given agree with common practice with apparatus of similar capacities and speeds and cannot be questioned. The practicability of the generator efficiency proposed can be proved by comparison with actual test results on a generating unit rated 3,000 kilowatts, of which many are in use, and which have been repeatedly investigated with the greatest thoroughness. In almost all turbine units increase of capacity is decidedly advantageous to economy, and such advantage would exist in this case, although the efficiencies assumed are not superior to those shown by these actual test results.

Advantages.—In this plan for electric propulsion the electrical apparatus simply serves as a speed-reducing bond between the turbine and the propellers, and it may be asked why electricity should be used when the practicability of other methods of speed reduction have been asserted, and to some extent verified, by experiments.

The answers to this are:

First. Electricity is capable of efficiently effecting reduction in a very large ratio, the reduction in this case being in the ratio of 50 : 6.

Second. That the electric speed reduction is susceptible to change of ratio, which makes possible efficient action at different speeds.

Third. The electric speed change involves no kind of complication, difficulty or uncertainty.

The efficiency of this electrical bond will be about 92 percent at all speeds, and this will remain constant through the life of the apparatus. It is questionable whether any other practicable form of speed reduction in such a ratio can be made equally efficient when all friction losses are considered, even if it should be proved that other methods are practicable at all for use in such large ships. The particular type of turbine proposed in this case affords the great advantage of good efficiencies throughout wide ranges of load and speed, which characteristics in combination with the pole changing in motors are of great value in such a case.

The possibilities outlined in this paper, if true, are certainly of great importance to the shipbuilding industry, since they open a field which is almost entirely new. The most important existing electric drive installations of which the writer has knowledge are those of two fireboats in the city of Chicago which were equipped under his direction. While these contain small and relatively inefficient turbines and electrical apparatus, their performance from the first has been efficient, simple and entirely free from trouble. While the present paper relates alone to the propulsion of certain battleships, the figures and facts which it presents are fairly illustrative of a wide range of possibilities in the propulsion of vessels; and while other cases have not been specifically investigated, it is thought that there are a very large number in which electric drive would be better, simpler and cheaper than anything heretofore produced.

NEW SWEDISH AND DANISH SUBMERSIBLES.

BY ROBERT G. SKERRETT.

Fig. 1 shows the *Hvalen* during her official trials at Spezia preliminary to her acceptance and the long run to Stockholm. She is of 230 tons submerged displacement, and is propelled by three screws. During her trials she developed a maximum surface speed of 15.2 knots—her Fiat motors producing 1,050 horsepower. Submerged, the boat makes 7 knots.

The Swedish authorities have become seriously convinced that submarine vessels promise a very effective defnse for the seaboard and the principal waterways of the kingdom, and the building of the *Hvalen* is the first step in the execution of a programme which calls for an extensive flotilla of this order of fighting craft. This is not the first effort that Sweden has made in this direction. It was due to native enterprise that the Nordenfeldt submarines first gained recognition and gave promise of practically solving the difficulties of submarine navigation in the "eighties," but it was not until the advent of the Holland boat that the Swedish admiralty gave any substantial encouragement to the art, when it built the *Hajen,* a virtual reproduction of the *Adder* class of the United States navy.

In 1907, however, the Ministry of Marine, after much painstaking investigation and deliberation, invited proposals from the principal commercial builders of under-water craft. In the competition which followed, designs were offered by Vickers Sons & Maxim, of England; Laubeuf, of France; Krupp, of Germany; Lake, of America, and Laurenti, of the Fiat-San Giorgio, of Italy. The choice fell to the Italian boat, and the *Hvalen* is the consequence.

One of the contract conditions was that the *Hvalen* should

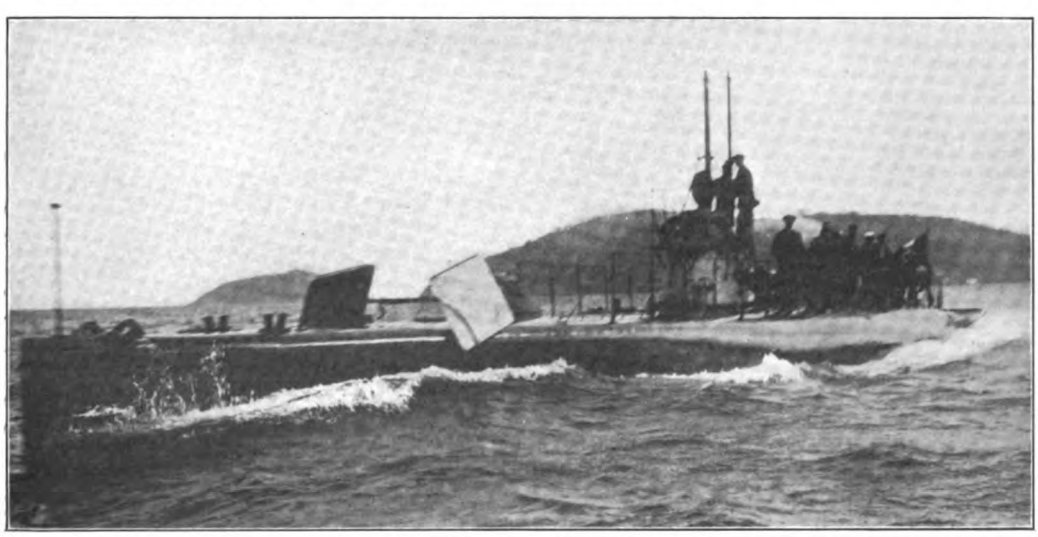

FIG. 1.—THE HVALEN UNDERGOING HER OFFICIAL TRIALS AT SPEZIA.

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demonstrate her sea-keeping qualities by making the extraordinary run of 4,000 nautical miles from Spezia, Italy, to Stockholm, Sweden. She was obliged to be self-sustaining, and to cover the distance absolutely unattended by either an escort or mother ship.

The first leg of the journey was from Spezia to Cartagena, Spain, a distance of 790 nautical miles. The boat made this run under her single central screw, and covered the distance in seventy-two hours, an average speed of 10.97 knots, the two wing screws being disconnected and allowed to revolve freely. During this run the boat met with heavy weather, and yet she went on without halt. Her commanding officer, a lieutenant in the Swedish navy, reports that the boat showed remarkable seaworthiness, and the crew arrived at Cartagena in excellent condition.

From Cartagena the *Hvalen* went to Gibraltar, and from Gibraltar to Lisbon. During the run to Lisbon the boat used her twin screws while working her way in the teeth of a regular easterly gale, and, notwithstanding the blow, made an average speed of 11.5 knots. This is a striking performance when we recognize that the best American submarines have made but 11 knots over a measured-mile course and under very favorable weather conditions. The other stages of the run were as follows: From Lisbon to Vigo, from Vigo to Ferrol, from Ferrol to Brest, from Brest to Portsmouth, from Portsmouth to Kiel, and from Kiel to Stockholm.

The principal dimensions of the *Hvalen* are:

Length over all	140 feet.
Beam, maximum	14 feet.
Displacement, when fully submerged	230 tons.
Displacement, light	186 tons.
Number of screws	3
Number of screws used when submerged	2
Maximum brake-horsepower	1,050
Maximum surface speed	15.2 knots.
Radius of action at full speed	480 nautical miles.
Radius of action at 10-knot cruising speed	1,000 nautical miles.
Maximum submerged speed	7 knots.

The boat has a reserve buoyancy normally on the surface of 24 percent of her light displacement, and by a special arrangement of the superstructure it is possible to increase this to 60 percent, thus giving the vessels of the *Laurenti* type a surface reserve of buoyancy fully as great as that of the ordinary above-water torpedo boat. Because of the special features used in connection with this superstructure, this unusual reserve buoyancy does not add any impediment to the rapid submergence of the boat or her emergence and return to the light cruising condition from an under-water run. The superstructure fills automatically when submerging, and is self-bailing when emerging, so there is no occasion for the installation of heavy pumps or the use of any power to control this mass of water

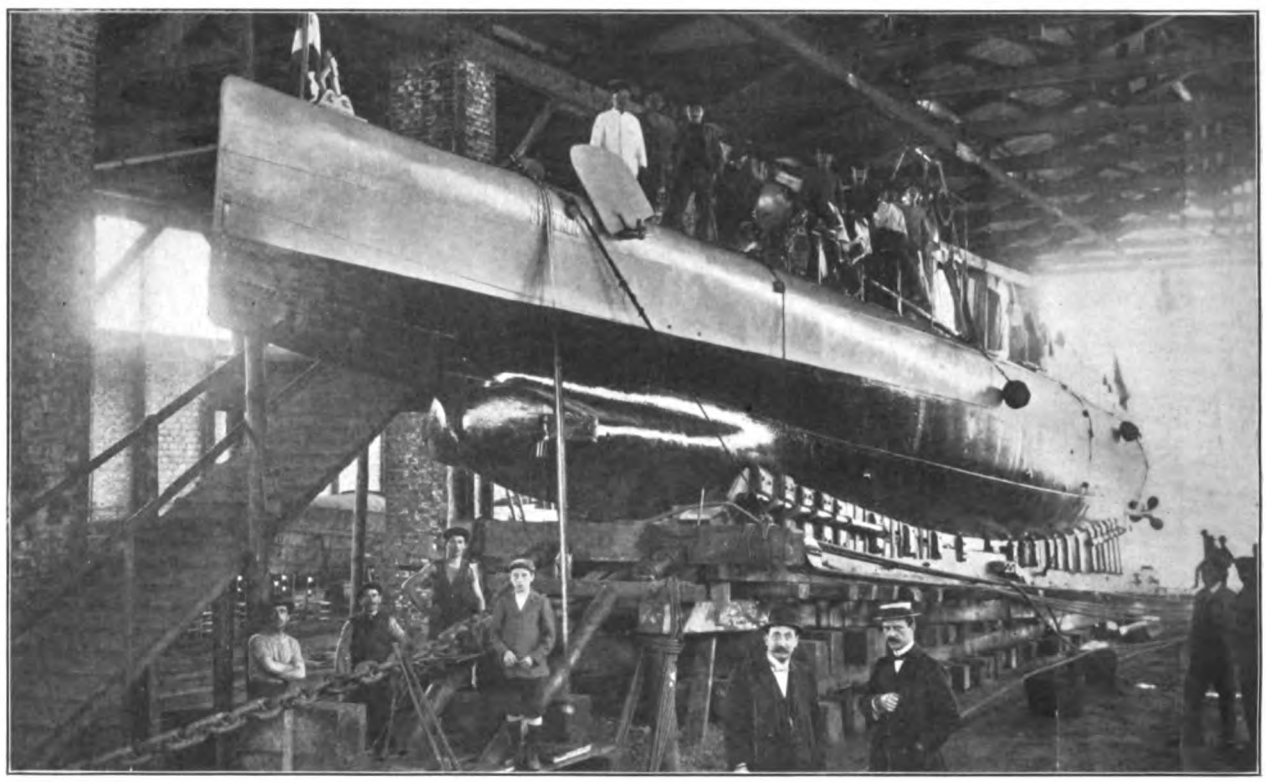

FIG. 2.—DANISH SUBMERSIBLE DYKKEREN.

In America, interest naturally centers in this boat, because an enlarged edition of the type is now building at the yards of Messrs. William Cramp & Sons Company for the United States navy.

The Danish Government, like its neighbor, has embarked upon a programme of submarine construction, and has built a small submersible of the *Laurenti* type. This boat is named *Dykkeren*, and has a modest submerged displacement of 128 tons; in this particular being a virtual counterpart of the American submarines of the *Moccasin* class. The building of the *Dykkeren* constitutes a record performance. The keel of the boat was laid on Sept. 9, 1908, she was launched on July 18, 1909, and finished her final delivery trials Aug. 20, 1909. The *Dykkeren* is entirely electrically propelled, and was designed to meet special conditions prescribed by the Danish Admiralty. Any one studying the coastline of Denmark will readily appreciate that for purely defensive work a submarine vessel need not have a great radius of action.

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PERFORMANCE OF DANISH SUBMERSIBLE DYKKEREN.

	Contract Requirements.	Actual Performance.
Maximum surface speed, knots.....	11.0	12.02
Cruising speed, knots.....	7.0	8.1
Submerged speed, knots.....	7.25	7.5
Radius of action at full speed, nautical miles	18.5	24.0

The time required to pass from complete buoyancy to complete submergence is something less than five minutes, and to return to the surface and be under way at cruising speed requires only three minutes. The soundness of the hull was demonstrated by subjecting the boat to hydrostatic pressure at a depth of 145 feet, when there was no sign of deformation.

The *Dykkeren* is 113 feet 8 inches long, and has a maximum beam of nearly 11 feet. On the surface her displacement is 103 tons, and when submerged 128 tons. The photograph of this boat is of interest because of the installation of the two bow-torpedo tubes. As one can see, these tubes lie well below the waterline and back from the stem of the vessel. Apart from the protection thus afforded to the tubes in case of a bow-on collision, their position insures an equalizing of lateral pressures at the instant the torpedo is discharged, so there is less danger of deflection due to these causes and a minimum of risk of wrenching the tail of the torpedo before it is clear of the tube. Another feature of interest shown by this photograph is the housing of the submerging rudders, so as to get them out of the way of the sea when running on the surface. Where these rudders extend rigidly outboard from the sides of the vessel they are liable to damage when coming alongside a dock or to be strained and bent by the pounding of a seaway.

THE MARINE STEAM ENGINE INDICATOR—V.*

BY LIEUT. CHARLES S. ROOT, U. S. R. C. S.

The American Thompson instrument, as made by the American Steam Gauge & Valve Manufacturing Company, is shown in Fig. 36. This is a good example of the modern instrument with an outside spring. The pencil mechanism is practically the same as that shown in Fig. 28, and has the shortened front link. The detent motion is of a type peculiar to this make of instrument. By means of the lever, whose handle is just above the 1899 patent date, the drum and its base-ring may be thrown out of gear. By this means the drum is made stationary at will, for the inspection or changing of cards, while the cord remains taut and in motion with the engine as usual.

Another form of instrument with the shortened front link is shown in Fig. 37. This is one of the many patterns made by the Star Brass Manufacturing Company, of Boston, Mass. It differs from most other instruments, in that it takes steam above instead of below the piston, and its piston area is but ¼ of a square inch. By means of this smaller piston lighter springs are used, and the weight of the reciprocating

* Copyright, 1909, by Charles S. Root.

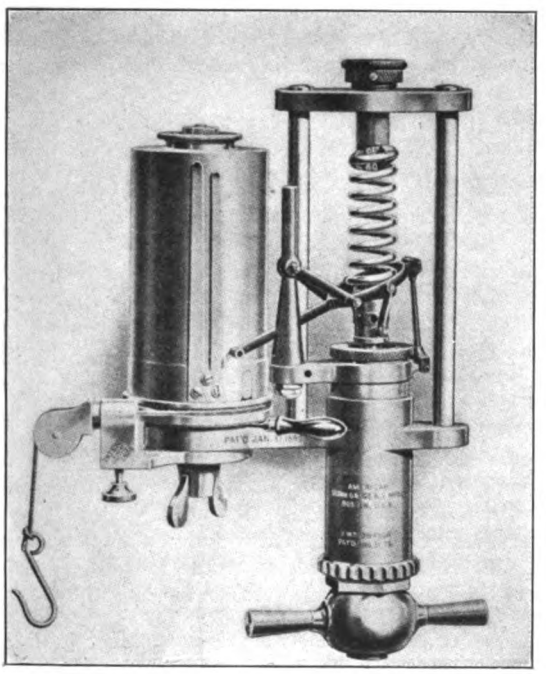

FIG. 36.

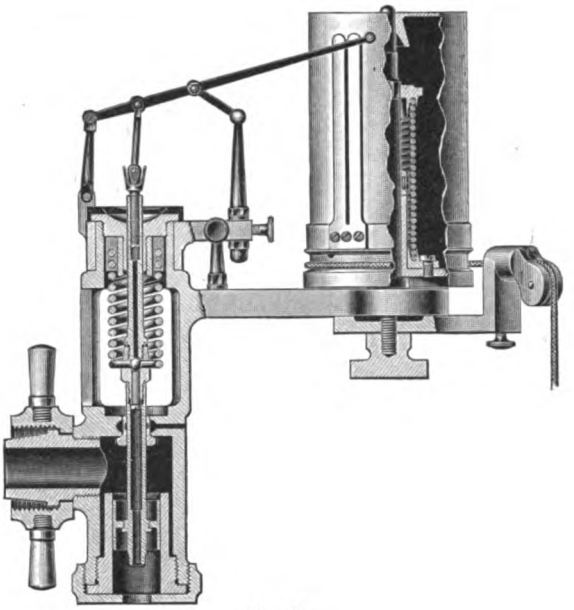

FIG. 37.

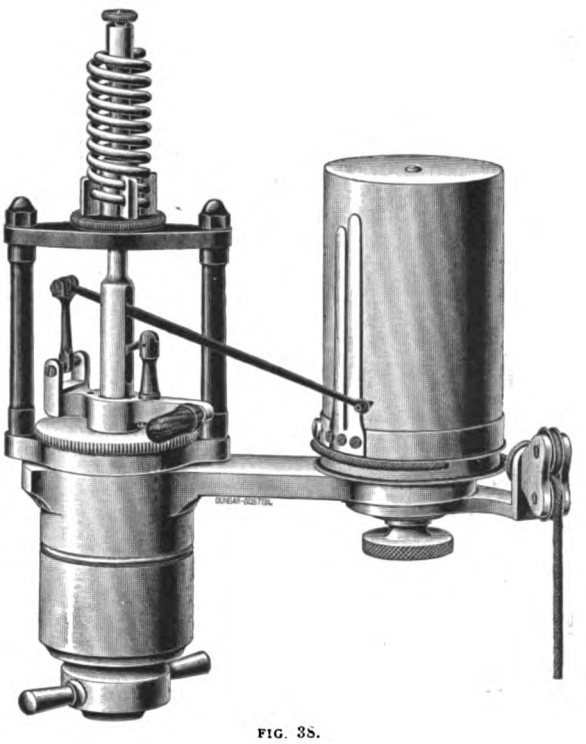

FIG. 38.

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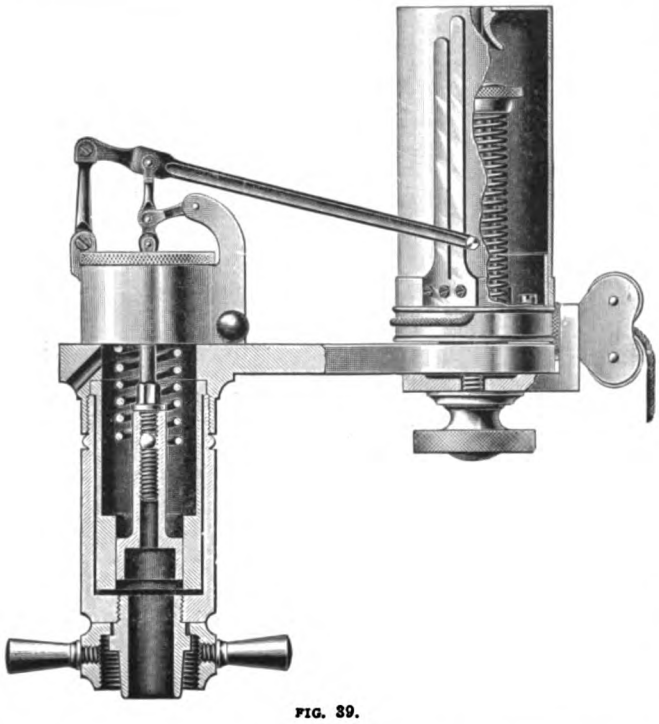

FIG. 39.

parts is no greater than in the same make of instrument with an inside spring and a ½-inch area piston.

In Fig. 38 is exhibited the Crosby indicator, with outside spring. The pencil mechanism of this instrument has its front link attached to the piston-rod connecting link, and has a steam piston of 1 square inch in area, made in the form of the central zone of a sphere. Fig. 39 shows an instrument of the same make, suitable for use either as a steam or gas engine indicator. It is furnished with one piston of ½ inch area for use with steam engines, and another piston of ¼ inch area for gas engines.

A Crosby instrument, as made for ordinance and hydraulic work, is shown in Fig. 40. The pencil mechanism has a post, bearing lightly against the pencil arm, to keep the scriber in

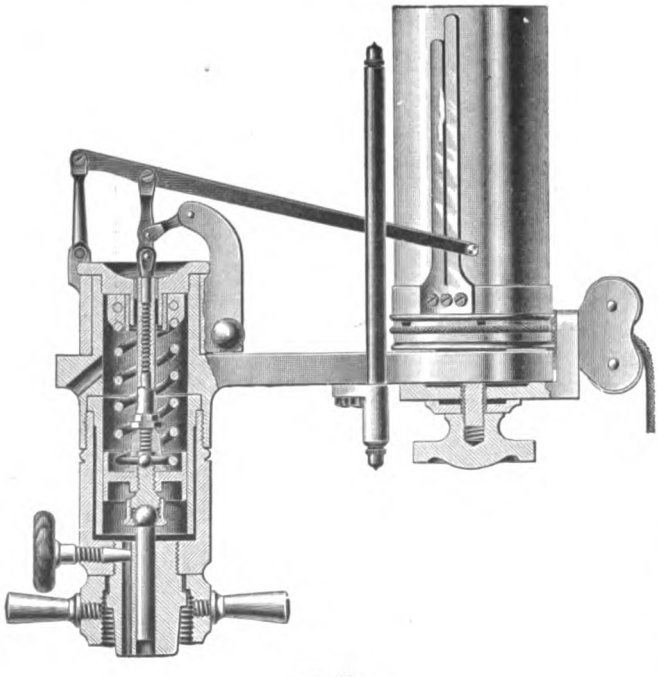

FIG. 40.

contact with the card during sudden shocks. The lower piston, or, more properly, the plunger, is of very small area, and is fitted in a bored hole in the union casting at the bottom of the instrument. When moderate pressures are to be indicated the by-pass valve at the left is opened and the main piston comes into play.

In the Tabor instrument, fitted with an outside spring and an electrical attachment for rotating the swivel head, the pencil motion is rectified by a sliding pair, whose path is curved, as shown in detail in Fig. 41. The instrument, as

FIG. 41.

shown, is the product of the Ashcroft Manufacturing Company. The flat spring indicator of Batchelder is shown in Fig. 42. The spring is at *B*, and its "scale" is altered by shifting the slide and fulcrum *S* to the right or left. The pencil is guided by a straight slide, as shown in Fig. 30.

Many other forms of indicator, additional to those described here, have been designed and made, but only one other kind will be noticed.

For extremely high speeds, the ordinary indicator will not answer, owing to the disturbing effects of inertia. Some years ago Mr. Carpentier, of Paris, France, designed an instrument which he called a Manographie. He arranged a small mirror so that it was deflected in one direction by changes in cylinder

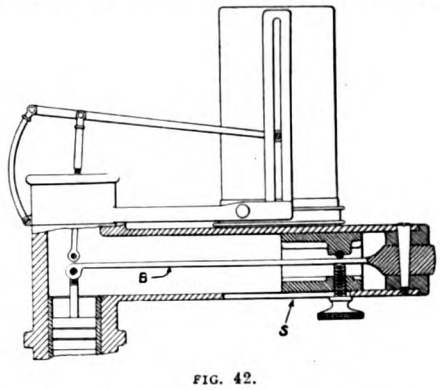

FIG. 42.

pressure, and in a direction at right angles by the movement of the engine cross-head. A lamp was so located that a beam of light thrown on the mirror was reflected in a point on a ground-glass screen or photographic plate, and a momentary view or a permanent record of the diagram was obtained. This apparatus was successfully used on gas engines running at speeds of 2,000 revolutions per minute.

In concluding this section, the attention of the reader is called to the fact that most indicator manufacturers are in a position to furnish instruments with large or small pistons, inside or outside springs, large or small drums or those fitted for continuous cards, instruments with but one size of piston or the combination of large and small cylinders, and, lastly, instruments entirely steel fitted for use on ammonia compressors. The user is thus free to choose the instrument whose general design most nearly satisfies his idea of what an indicator should be.

For high-speed work lightness of the moving parts must be the prime consideration, but for moderate or slow speeds this

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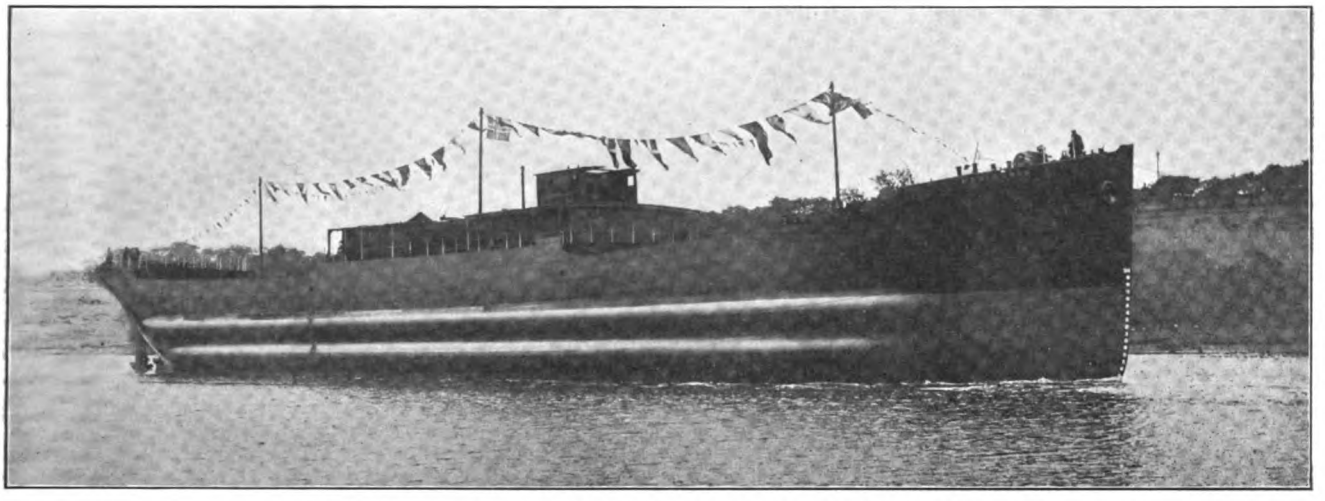
may be sacrificed for other advantages which the purchaser may desire. Consideration should also be given to the ease with which springs may be changed, lost motion taken up and the instruments changed from right to left hand.



S. S. MONITORIA IMMEDIATELY AFTER LAUNCHING, SHOWING EXTENT OF CORRUGATIONS.

THE MONITORIA.

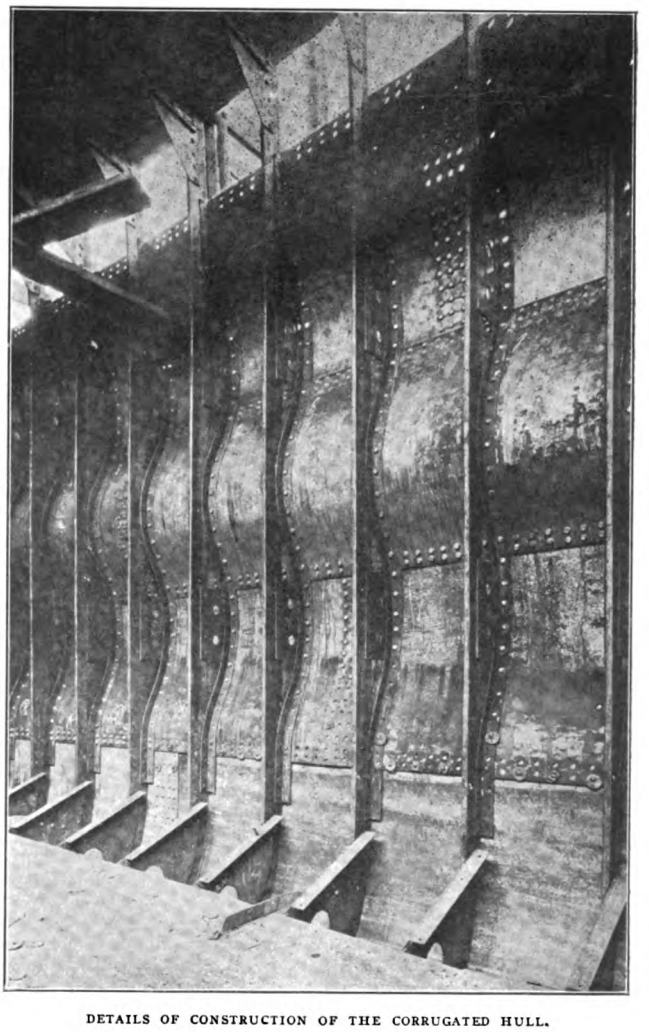
BY ARTHUR H. HAVER.

The latest improvement in the construction of ships consists in an application to the outside form. This part of the ship has hitherto been carefully ignored by the majority of ship designers, as it has evidently been concluded that the ordinary ship form did not admit of any radical improvement in reducing resistance. It is, therefore, more than usually interesting to know that the *Monitoria,* an ordinary cargo steamer, 279 feet 6 inches long, 39 feet 10½ inches breadth, molded, and 20 feet 7½ inches depth, molded, of unusual form, has proved to possess remarkable advantages, which indicate that a law hitherto ignored has been applied which enhances the value and improves the speed of ships, or what is the same thing, reduces the horsepower and coal consumption of all classes of vessels.

Instead of the usual wall-sided shell plating, this vessel has been designed with two longitudinal projections along the outside of the ship between the load waterline and the bilge, in the form of sections of rather flat arcs, so that the breadth of the ship becomes nearly 42 feet extreme, or about 22 inches wider than the molded breadth. These corrugations do not extend to the extreme ends of the vessel, but stop where the hull begins to fine forward and aft. Naturally, these projections give more buoyancy and add to the displacement of the vessel; they also increase the wetted surface and increase the periphery of the shell plating girths. Although these conditions must be considered as prejudicial to speed, nevertheless in this ship greater speed was proved to exist, both by numerous tank experiments extending over nearly four years and by results on the measured mile and on ocean voyages.

The improvement of 5 percent in speed at 10 knots disclosed by the tank experiments was realized on the actual ship, while at 9 knots instead of the 4.16 percent greater speed anticipated the actual increase proved to be 4.23 percent.

This vessel is owned by the Ericsson Shipping Company, of Newcastle-on-Tyne, and was built by Messrs. Osbourne, Grahame & Company, of Hylton, Sunderland. This firm has made a specialty of a type of vessel of the ordinary three-island construction, to carry 3,200 tons dead weight, cargo and bunkers, and the special form, tonnage, speed and power of these vessels have proved so economical that twenty-three vessels have been built from the same design practically without alteration. Owners of these vessels have expressed themselves satisfied with their good performance, one remarking that if anyone could improve the earning power of his ship it must be a noteworthy achievement. This type of vessel was therefore chosen as the one on which to try these longitudinal projections, principally from the fact that it was a successful design, and not one remarkable for inefficiency.



DETAILS OF CONSTRUCTION OF THE CORRUGATED HULL.

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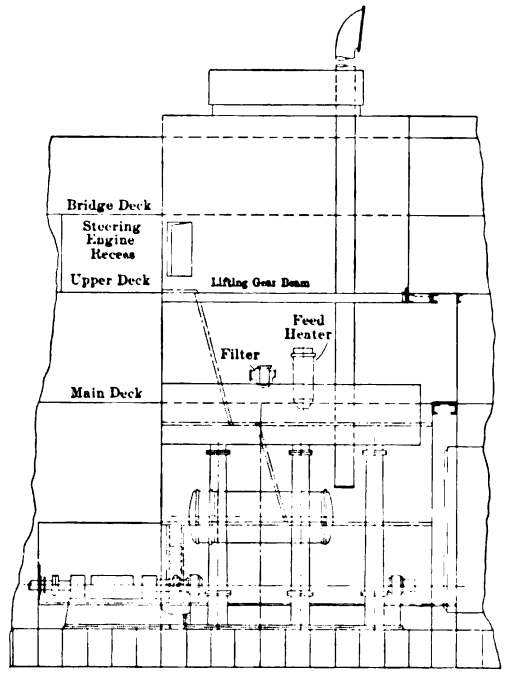
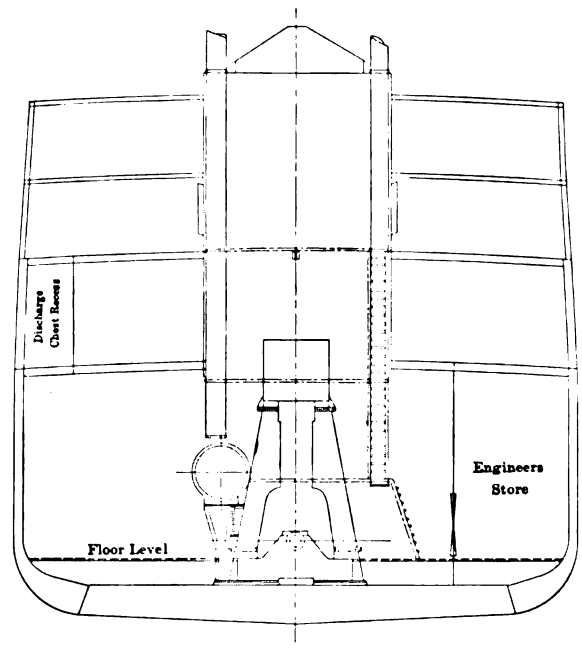
The trial under ballast conditions, with the corrugations above the ballast waterline, proved that the *Monitoria* was no different from any of the sister ships, the same speed and power being obtained. The loaded trial, however, showed that when the corrugations were immersed, practically the same speed was attained as when light, although at this draft she carried about 135 tons more displacement than her sister vessels in loaded condition, and with the same engines, indicated horsepower and propeller the speed was three-eighths

	A.	B.	Monitoria			
Trial displacement....	4,440	4,450	4,575	4,575	4,575	4,575
Knots speed	9.76	9.78	9.68	9.78	9.96	10.12
Indicated horsepower..	1,133	1,116	966	1,000	1,120	1,195
Revolutions	70½	70	65½	66¼	68¼	69¼
Draft	17.8	17.8½	18.0	18.0	18.0	18.0
Steam pressure	180	180	175	175	175	175

Propeller slip from 3 percent to 6 percent.

A noticeable feature was the steadiness of the *Monitoria's* motion through the water. There was no vibration, and no broken water was to be seen leaping up and running freely





Engine Room, Looking Forward

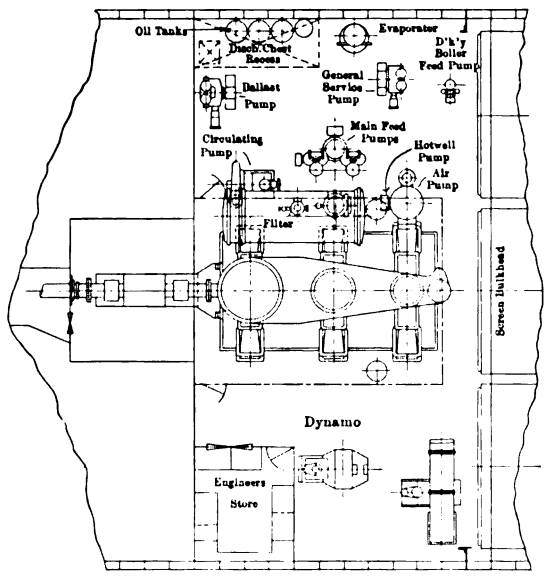


FIG. 10.

of a knot more than that of her sister ships, which means that at the same speed as her sister ships she was capable of being propelled with 14 percent less power.

The engines have cylinders 21, 33, 56 inches by 36 inches stroke; there are two boilers, operating at 180 pounds pressure, with a heating surface of 3,000 square feet.

In the following table two of the sister ships are marked A and B; the *Monitoria* was run on the measured mile, progressively. The results, compared with vessels A and B, are as follows:

along the sides; but, instead, a smooth, subdued, undulating wave line about 3 feet wide took its place, perfectly clear and transparent, so that the projections on the hull 10 feet below were quite easily discernible. The usual foam-covered surface, caused by the broken head wave, was carried about 3 feet clear of the ship's side and parallel to it, showing that a most noteworthy subduing action of the stream-line waves was operating for the ship's benefit. The increased speed (or it can be expressed as a reduction of horsepower) appears to have been attained by robbing a portion of the power usually expended in creating the amplitude or height of the waves and at the same time increasing the wave length due to the restricted amplitude. The power saved in this vertical restriction of wave becomes available for horizontal propulsion. The increased length of wave gives a greater speed of wave, which reduces the friction power, eddy formation and bow wave, besides assisting all other stream lines around the ship.

Naturally, as there is a greater frictional area, due to the greater periphery of the shell, there must be more frictional resistance. The amount of power to be allotted to this frictional resistance may be open to doubt, and many text books could be proved to be wrong in the amount they allot to it, but whatever the amount lost by the increase of wetted surface, due to the corrugations, there is no doubt whatever that more power is available, not only to overcome this increase of wetted area but also additional power for the propulsion of the ship.

In the type of vessel, of which the *Monitoria* is the first, we have a vessel which actually accomplishes the hitherto impossible task of carrying 3 percent more cargo in a hull having over 3 percent more displacement, yet doing it at her commercial speed with over 14 percent less indicated horsepower.

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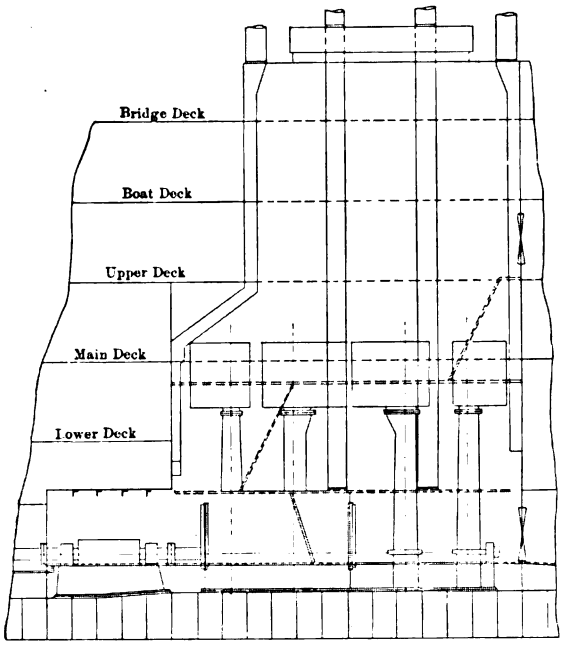
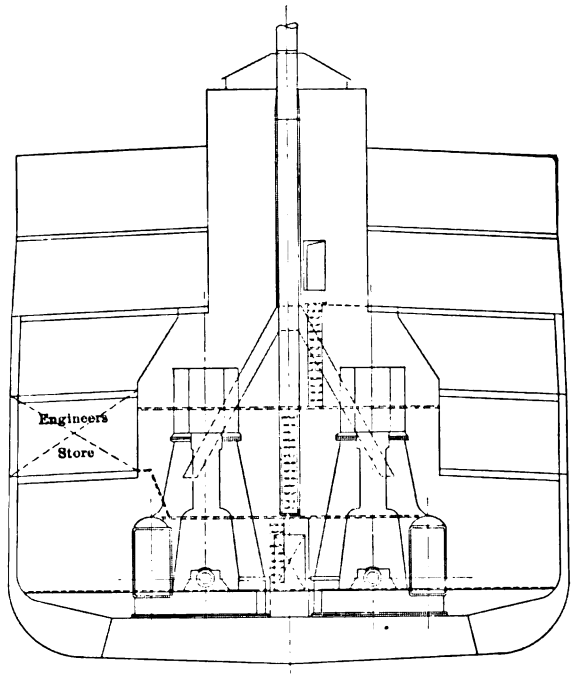
The size, proportions and position of these corrugations must be decided upon to suit the form, fineness, speed and size of the vessel. Varying proportions give varying results. Full





Section looking Forward

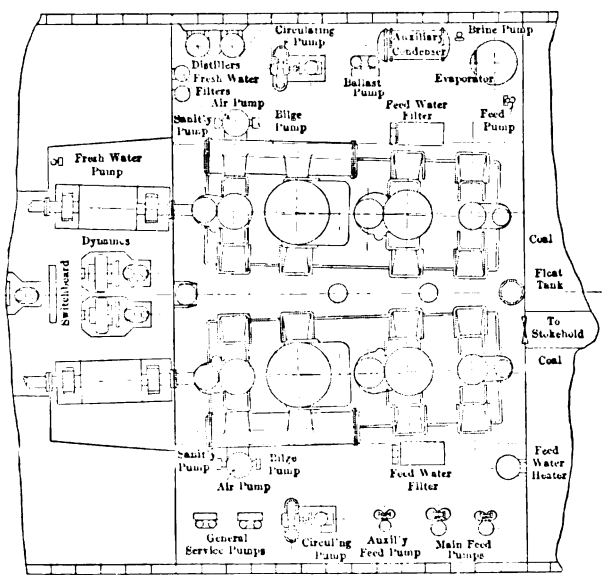


FIG. 11.

ships give greater percentages of improvement than very fine ones. In respect to rolling, the corrugations act in the same manner as a bilge keel, causing the ship to have a slower period of roll besides effecting a reduction in the amplitude of the roll. This also occurs longitudinally by crushing the end waves which would tend to rise on deck. These points make for a better sea boat, fewer shocks, greater steadiness and greater strength. The action of the corrugations in reducing the eddy at the after end causes the oblique stream lines to be more nearly horizontal, giving a better run of water to the propeller and rudder, resulting in better steering and less slip of the propeller. The ship can also make a better passage in a head sea, and has more strength to resist the hogging and sagging strains.

A battleship or cruiser fitted with these corrugations would have a steadier gun platform, a greater range of action, carry more fighting metal, and have a greater speed. A passenger boat would be steadier, stronger and more comfortable, be more economical in coal and carry more weight.

MACHINERY AND PIPING ARRANGEMENTS ON BOARD SHIP—II.

BY JOHN M'COLL.

THE ENGINE ROOM.

In fixing the position of the main engine for a single-screw vessel, much will depend on whether the shaft is required to be parallel to the base line, or whether it may be raked. It often happens that with an average diameter of propeller, ordinary depth of sole plate, and depth of ship floors, the distance from the base line to shaft center is much less in the engine room than at the stern frame. If the rake of the shaft is small, the aft end of the sole plate is made deeper, to keep the holding-down flange parallel with the tank top. If the shaft must be parallel to the base line, two methods may be adopted to make up the level in the engine room. A complete seat of girders and tie plates may be built on the tank top or the whole tank in way of the engine may be increased in depth. The latter is the better way and is that usually adopted.

It is important that the holding-down bolts should, if possible, pass through the floor angles, in addition to the tank-top plating. This can be arranged by fitting broad angles at the top of the floors, or broader flanges at the bottom of the sole plate, or, inclining the floor at the top, forward or aft, as the case requires. A tracing of the bottom of the sole plate should be made and laid on top of the tank-plating drawing; this moved forward or aft from the approximate position will enable a compromise to be readily made.

The thickness of the tank-top plate under the main engines is increased to from ¾ inch to 1 inch thick; and care should be taken that the fore-and-aft laps do not hinder the proper fitting of chocks all round. The thickness of chocks should not exceed 2 inches, and each holding-down bolt should pass through a cast iron chock.

Assuming, in the first case, that the shaft is not to be raked, and the position of the boilers has been fixed, the arrangement may be as that shown in Fig. 10. At the forward end suffi-

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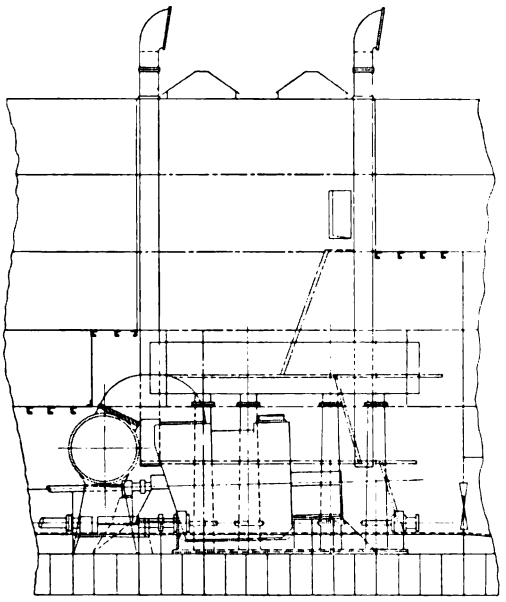
cient space must be allowed for a passage from side to side. If the high-pressure valve is at the forward end of the cylinders, then the eccentrics at the floor level, and quadrants and wiper shaft lever higher up, must be considered. A guard will be required around the eccentrics; and the boiler mountings on the back of boilers, especially in jobs with three boilers abreast, may project into the apparent clear space and block the passage. At the aft end it is advisable to provide a passage across the engine room, and this is usually done by placing a bridge over the shaft between the sole plate and thrust block.



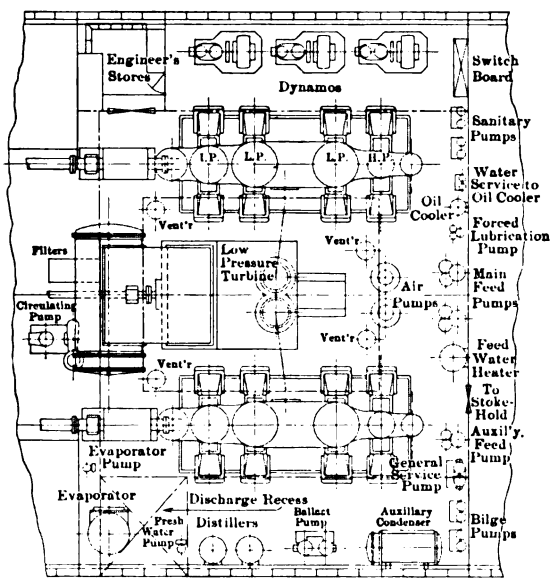
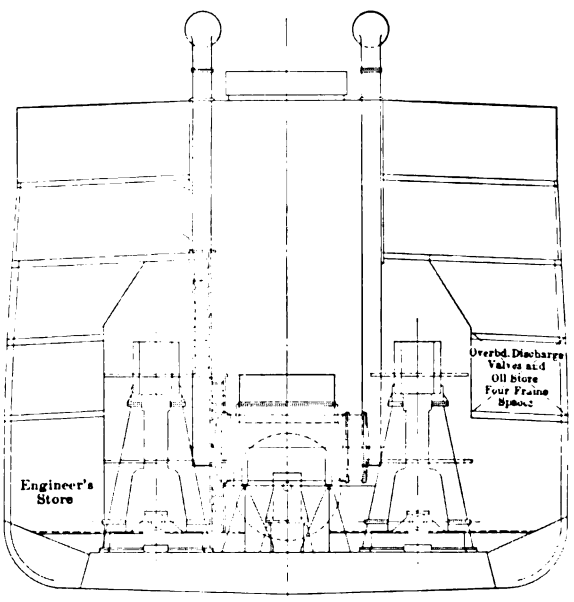


FIG. 12.

If for any reason the condenser tubes cannot be withdrawn from the forward end, provision must be made for that purpose at the aft end. Two methods may be used to meet this requirement. A hole in line with the condenser may be cut in the aft bulkhead and fitted with a portable plate, or the thrust block recess may be made wide enough to suit, as shown in Fig. 11. In the former case it may not be convenient when required to remove the plate and shift whatever cargo may be behind it. In the latter case there need be no delay, and many uses could be found for the little extra space taken up. In using the portable plate method, the thrust recess side may be so close to the block that it prevents chocks from being properly fitted there; in that case the plates on that side should be left loose till the block is bolted down. On the working side of the thrust block the width of recess is fixed by the width of the watertight door to the tunnel, and, as it is seldom possible to haul a vertical sliding door, space for a horizontal door has to be allowed. The height of the recess need not be more than gives ample head room entering from the engine room.



ENGINE ROOM LOOKING FORW.

The casings in way of cylinders should be wide enough to have a passage all round, more so if the receiver pipes project above the top grating. In fixing the casings above the cylinders, the points to be considered are: the cylinder-cover lifting gear, the entrance ladder or ladders, and the ventilation. In small jobs a rough guide for width of casings is that it should be sufficient to let the sole plate down without having loose work.

Generally there should be as little loose work as possible, and that is again a matter for arrangement between the shipbuilder and the engineer. Should the fitting out take place in a side way, liable to be disturbed by passing traffic, it is obvious that more space for shipping engines and boilers should be allowed than in the case where a ship may lie in a private basin and be free from such disturbing influences. In special cases it will be advisable to consult with the engineer in charge of the fitting out regarding his proposals for shipping the engine and boilers, as he should know best the capacities of his cranes and other appliances. In most cases, and especially in ships of light scantlings, a shell plate low down in way of the engine room should be left off till near the launch for the shipping of shafts, pumps and other gear. If the overboard discharge valves are required to be above the load waterline, space along the ship side will be taken for these and access to the space will also be required. Should the space be conveniently accessible, it is usually fitted up as a store for heavy tackle. If, however, these valves may be below the load waterline, no special provision need be made for them. If asked for early, shell butt straps or laps may be arranged by the shipbuilders, to suit the best positions for the larger valves.

In twin-screw jobs, the distance between engines depends mainly on what is considered a reasonable working space. It varies from 18 inches for small to 46 inches for large engines, measuring at the bottom flanges of the front columns at the forward end. As the types of engines and types of ships vary so much, the distance has to be arranged to suit each case.

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Fig. 11 shows an arrangement for a high-class twin-screw steamer where a fair compromise has been arrived at between shipbuilders and engineers regarding the space taken up and simplicity in design of casings. In this connection, while the engineer is upholding his requirements for working room, the shipbuilders' work has to be considered, and the more simple the casings are the better for all concerned. As the work proceeds the engineer should see that he gets early information about all pillars, stiffeners and bracket plates, so that he may keep clear of these with his fittings.

Fig. 12 shows an arrangement for a combination of two reciprocating engines with one low-pressure turbine. The reciprocating engines, as usual, rest on the tank top. The height of the turbine is fixed by the distance between the foot valves of the air pumps and the lowest part of the turbine blading. This distance varies from 16 inches to 26 inches, to insure the efficient draining of the turbine. The position fore and aft may be fixed roughly by keeping the steam inlet on the turbine in line with the exhaust branch on the reciprocating engines. If placed much further forward the engine centers are made wider, and little space is left for a convenient arrangement of auxiliaries in way of the starting platform. If placed further aft, more room is taken up for the condenser. The condenser may be made in two parts, and placed low down between the shafts, with the tubes fore and aft; but if made in one it should ride over the center shaft, close up to the turbine, with the tubes across the ship. The rake of the shafts, both horizontal and vertical, depends in each case on the shape of the stern and the diameter of the propellers.

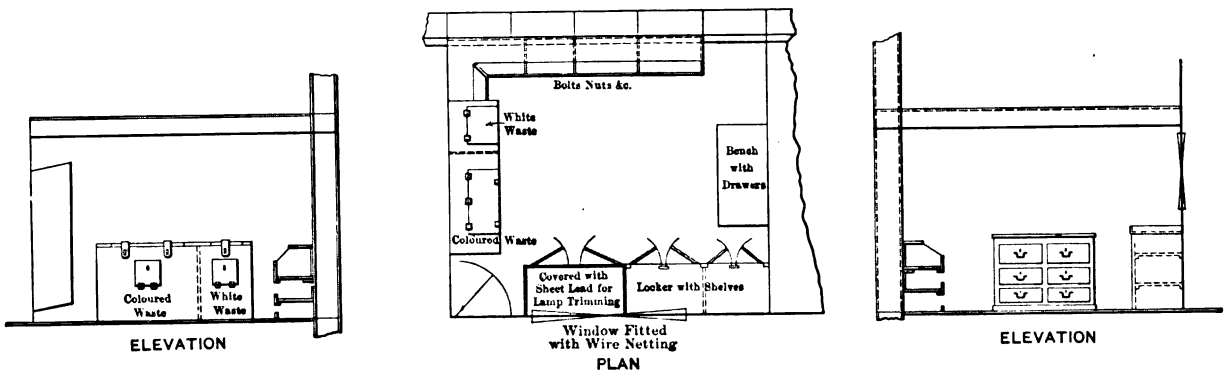

FIG. 13.—ARRANGEMENT OF ENGINEER'S STOREROOM.

AUXILIARIES.

A good arrangement of auxiliaries is of great importance, as on that will depend the handiness and simplicity of connections, the accessibility for repairs, the easy working of the engine room generally, and the first cost. They should be grouped in their proper relation and sequence, so that, say, the air pumps, filters (if of the gravitation type), feed pumps and feed-water heater are together, the evaporator, distiller, filters and fresh-water pump are conveniently placed, and the sanitary, ballast and bilge pumps are beside each other. The general service pump is usually placed about the forward end of the engine room, and the ash ejector pump, if in the engine room, should be as near the boiler room as possible. This arrangement of auxiliaries is not always possible; for, although it has become usual to have all auxiliaries independent, many firms still prefer to have at least the air, bilge and sanitary pumps worked off the main engines.

The electric generating machines are sometimes placed in the thrust-block recess, or on a flat at the aft end or at the side of the engine room. In some cases a smaller dynamo is placed high up in the casing, so that should the lower dynamos become flooded, some light may be obtained. If they are required to be in the engine room proper, they should be kept well above the engine-room floor level, and clear of any oil or water that may be thrown out by the main engines; but wherever situated they should always have the axis of the shaft lying fore and aft. It will be an advantage for the engineer in charge of the engine room to see whether all the usual auxiliaries are working correctly or not without leaving the starting platform. This can be done in some cases, and should be considered where convenient. As the refrigerating plant may be placed almost anywhere in a ship, the location of this can only be dealt with according to circumstances.

In designing seats for auxiliaries, consideration should be given to the fact as to whether the engines are fast or slow-running. In the former case, the seats have to be substantially and rigidly built. The top plates are made from $\frac{5}{8}$ inch to $\frac{7}{8}$ inch thick, and the side plates from $\frac{3}{8}$ inch to $\frac{5}{8}$ inch thick. In the latter case the top plates may be $\frac{1}{2}$ inch to $\frac{5}{8}$ inch thick, and the side plates $\frac{3}{8}$ inch thick. Large holes, and as many as possible without weakening the structure, should be made in the side plates; these will facilitate the building, and can be used for passing pipes through if required. Hardwood chocks, 1 inch thick, are fitted under auxiliaries, having large, flat bases; other auxiliaries should be bolted direct to the seat, and if thin chocks are needed these should be of steel. The seats should be so arranged, if possible, that the holding-down bolts pass through the top angles as well as the top plate.

The arrangement and size of the engineer's store room vary with the size and class of ship and the duration of voyage. It should be placed, if possible, about the floor level, and be well ventilated. If placed where, to reach it, ladders and gratings are required, these should be made wide, easy and strong, so that a man may carry heavy articles up or down without danger. An arrangement for an ocean-going steamer is shown in Fig. 13. Oil tanks are put in any convenient place about the engine room, usually on a side stringer, if at a suitable height. If the main engine cast iron columns are fitted as engine oil tanks, only small tanks will be required for the special oils, and these could be placed in the store. Round tanks are cheaper, but take up more space than those of rectangular section. The large tanks can be filled from deck with a portable leather hose, but permanent wrought iron piping can be arranged very simply, and is to be preferred.

The engine-room entrance ladders should be not less than 18 inches wide, with cast iron steps, and should have an easy slope. The top grating, in way of the cylinders, ought to be strong enough to bear a considerable weight, so that when overhauling, material may be safely laid there. All engine-room ladders should have cast iron steps, and, if possible, all gratings should have flat-topped spars.

In arranging floor plates, steps and inclines are to be avoided in the main passages, and plates ought not to be too

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large for one man to lift easily. Small plates imply more supports than large plates, but that is an advantage, as they are sometimes subjected to heavier burdens than they were expected to bear.

Intricate floor work around auxiliaries is not required, ready access to the fittings is the main end; but hand-rails, in way of the smaller auxiliaries, are sometimes an advantage, and give the engine room a better finished appearance.

THE NEW STEAMSHIP WILHELMINA.

The new steel steamship *Wilhelmina*, now building at the yards of the Newport News Shipbuilding & Dry Dock Company, Newport News, Va., for the Matson Navigation Company, of San Francisco, is intended for freight and passenger service between San Francisco and the Hawaiian Islands. She is of the following principal dimensions:

Length over all.....	451 feet.
Length on waterline.....	435 feet.
Beam, molded	54 feet.
Depth, molded to upper deck..	33 feet 6 inches.

The vessel is of the ocean-going type, with raised forecastle and combined bridge and poop, but differs from the general type of ocean-going vessels of similar dimensions and power in having her propelling machinery located in the after end of the vessel. She has the usual type of straight stem and elliptical stern, and is schooner-rigged with three steel pole masts. There are two complete steel decks in the hull, also an orlop deck in the forward hold. The vessel has a complete double bottom and a deep tank amidships, arranged to carry fuel oil or water ballast in all compartments except those under the engines and boilers, which are fitted to carry fresh water for boiler feed, etc. The space between the double bottom and orlop deck in No. 1 hold is also arranged to carry fuel oil or cargo. Above the bridge deck, amidships, are two tiers of steel houses, containing accommodations for passengers and the captain's quarters, and above these in a teak house are located the wheel-house and deck officers' quarters. A flying bridge is fitted at the level of the top of the upper house.

As before stated, the propelling machinery is located in the stern, the boilers and engine being in separate watertight compartments. The six main boilers are arranged with a fore and aft fire-room, all up-takes leading to a common stack. A vertical donkey boiler is located on the main deck level at the after end of the boiler room. Settling tanks for the fuel oil are located on each side of the vessel at the back of the boilers. The dynamo room is on the main deck abreast the engine room on the starboard side, the refrigerating plant on the port side, and the engineer's workshop at the after end.

HULL CONSTRUCTION.

The double bottom is built on the cellular type, with floor plates on each frame, a continuous vertical keel and two intercostal longitudinals on each side. The vertical keel is 60 inches deep throughout, oil-tight in all oil-carrying spaces and watertight elsewhere. All frames are cut at the margir plate, th latter being normal to the frames. Small expansion trunks, extending to the main deck, are fitted to all oil-carrying compartments, and dwarf cofferdams are fitted at all bulkheads enclosing oil spaces. Transverse framing is of the angle and reverse-bar type, with web frames spaced six spaces apart, and with side stringers of plate and angle construction. All deck beams are of channel section, supported by two continuous girders with wide-spaced cylindrical stanchions.

The orlop, main and upper decks are complete steel decks, the main being flush plated for convenience in trucking freight. Partial steel decks are also fitted on the bridge deck and upper bridge, these two latter decks also having complete calked decks of yellow pine with teak margin. Yellow pine calked decks are also laid on the forecastle, on top of all deck houses and in all passenger and crew accommodations on the upper deck.

The vessel is sub-divided into compartments by seven watertight bulkheads, all extending to the upper deck. All bulkheads in the accommodations aft are steel, as are also the machinery casings, which extend from the main deck to a full deck height above the bridge deck.

PASSENGER ACCOMMODATIONS.

Accommodations for first class passengers are all located amidships on the upper deck and in the bridge deck houses, there being forty-eight staterooms, with two berths each, and three special rooms with brass beds. All first class staterooms are fitted with metal berths, mahogany folding lavatories, and with a wide, upholstered transom seat which may also be used for a berth. Two special rooms on the bridge deck are paneled in black walnut, and all other passenger staterooms are paneled in pine, the ceiling in all staterooms being also paneled. The staterooms in the deck houses are all outside rooms, and those under the bridge are arranged generally in sections of four rooms each, with access from alcoves.

The main dining saloon, with a seating capacity for 140 persons, is located at the forward end of the bridge deck, and extends the full width of the vessel. It is paneled in white pine, and furnished with mahogany tables and sideboard and mahogany dining chairs with leather upholstered seats.

A large social hall is located at the forward end of the house on the bridge deck directly over the dining room. It is paneled in mahogany, and all seats are leather upholstered. The main stairways between the different decks in the passenger quarters are mahogany. The first class smoking room, located at the after end of the bridge deck house, is paneled in mahogany and has a beamed ceiling. A buffet is built in at the after end of the room. All upholstery in the smoking room is leather.

The first class toilet rooms are located on the upper deck at the after end of the first class quarters. These rooms are finished in mahogany, and have mosaic tile on the floors and a tiled wainscoting. Three bath-rooms for men and two for women are located in the toilet enclosures. In addition there are six single bath-rooms, four on the bridge deck and two on the upper bridge, access to which is obtained from the open deck and from the adjacent staterooms. All bath-rooms have tiled floors and tiled wainscoting.

The captain's quarters are located in the forward end of the upper bridge deck house, and consist of a stateroom and office, both paneled in mahogany. Direct access to the wheel house above is had by an inside stairway from the captain's office. Rooms for the first and for the second and third officers are located back of the wheel house in a deck house built of teak. The chief engineer has especially large quarters, consisting of a stateroom and separate office, finished in oak, and located aft of the engine hatch on the bridge deck. Quarters and mess rooms for other members of the ship's complement are located abreast of the machinery casings on the upper deck and ir the forecastle.

The space on the upper deck between first class quarters and the crew's accommodations is arranged to be used for cargo or for steerage passengers, there being portable metal berths provided for 108 of the latter.

The first class galley is located on the upper deck aft of the first class staterooms, and extends the full width of the vessel. Forward of the galley is the first class pantry, from which access is had to the passages leading to the dining room. All conveniences in use on modern, high-class vessels are fitted, including bake ovens, charcoal broiler, steam vegetable cooker, urns, steam table, egg boilers, etc. A "built-in" refrigerator is located at the forward end of the pantry. The ship's cold-storage rooms and room for stewards' stores are

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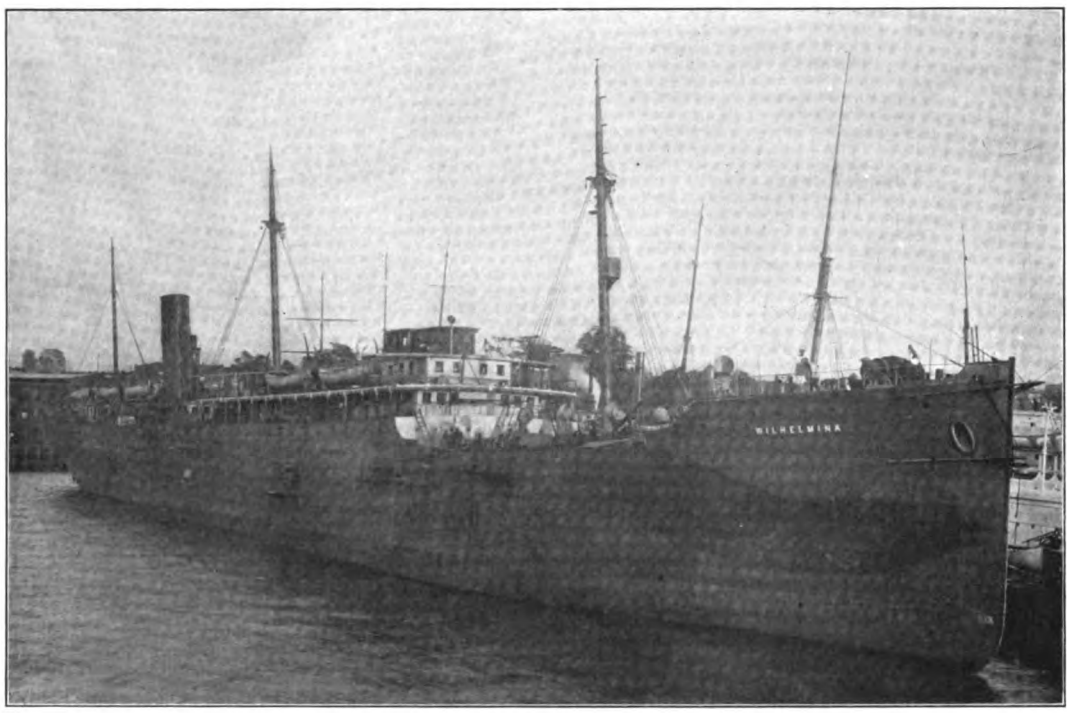


THE WILHELMINA FITTING OUT AT HER BUILDERS' DOCK.

located on the main deck just forward of the refrigerating plant.

CARGO ARRANGEMENT.

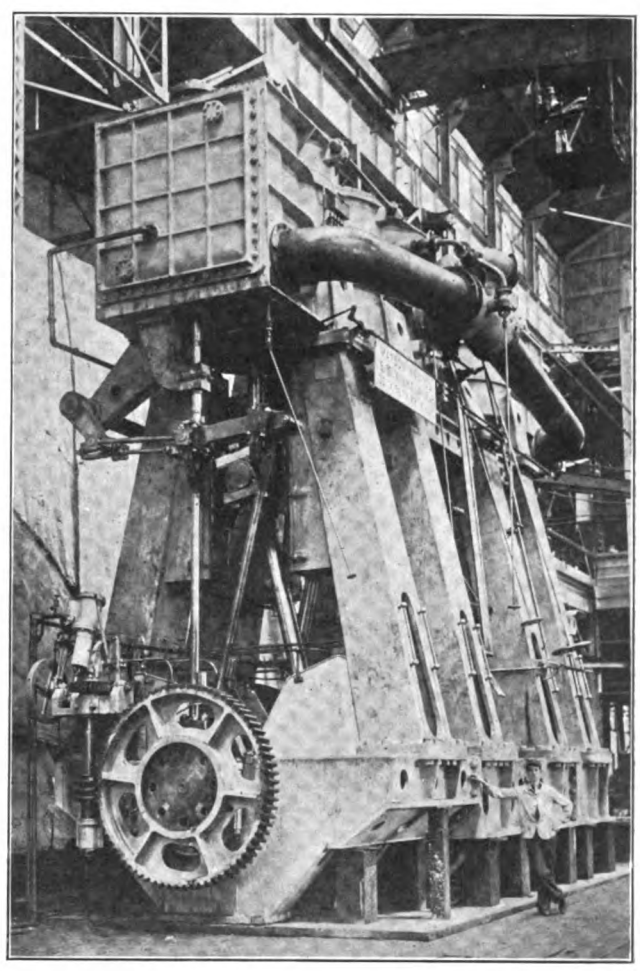
The cargo of the vessel will generally be more or less bulky in nature, and heavy machinery is also carried at times. To suit such conditions large hatches have been fitted and large cargo ports have also been provided, four of these being fitted on each side between the main and upper decks, and one on each side between the upper and bridge deck. All main cargo ports are fitted with watertight sliding doors. All cargo hatches are so arranged that they may be worked by booms attached to tables on the masts. For each of the main hatches there are two booms of 8 tons capacity each. There is also a 50-ton steel boom located on the after side of the foremast, and one 20-ton boom on the after side of the mainmast. An 8-ton boom is fitted on the forward side of the mizzen mast for handling engine-room weights.

For carrying refrigerated cargo, a cold-storage space of 10,450 cubic feet capacity is provided on the main deck amidships. The plant for circulating brine in this space consists of two 8-ton ammonia compression machines, supplied by the Vulcan Iron Works, of San Francisco.

The vessel is lighted throughout by electricity. The generating plant consists of three units of 5, 15 and 30-kilowatt capacity, respectively, all supplied by the General Electric Company. Other electrical items consist of a 7½-horsepower Richmond Electric Company motor for operating the workshop machinery, and a No. 6 Sturtevant blower for ventilating the upper 'tween deck space in No. 1 and No. 2 holds for the carrying of fruit. A 14-inch Rushmore searchlight is located on the top of the pilot house.

For handling the stockless anchors, which are of the Baldt type, there is a Hyde windlass, located on the forecastle deck, the engine for driving the windlass being located on the deck below. The latter engine also drives the capstan, which is located on the forecastle deck aft of the windlass.

The steam winches for handling the cargo consist of eight double 7 by 10-inch winches, supplied by Murray Bros., of San Francisco. There is also a Hyde double 8-inch by 10-inch



MAIN ENGINE OF THE WILHELMINA.

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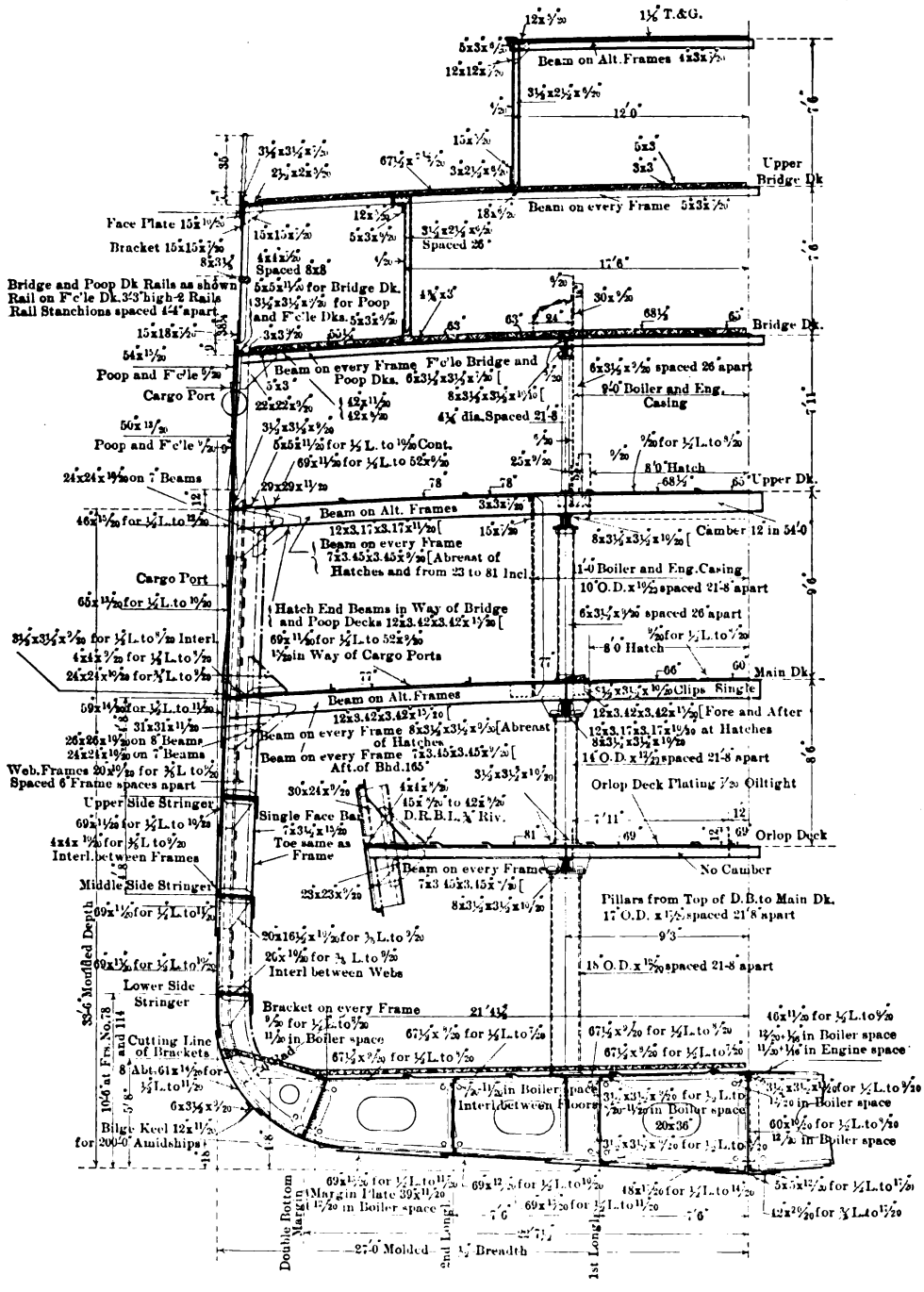
steam winch, located on the poop deck aft, for warping purposes.

The steam steering gear is of the Brown tiller type, supplied by the Hyde Windlass Company. It is operated from the pilot house by a telemotor, and an auxiliary steering stand is also attached directly to the gear.

Life-saving equipment is provided for 335 people. There are nine 28-foot metallic lifeboats and one 20-foot wood yawl, all boats being handled by Welin quadrant davits. The vessel is equipped with a Nicholson ship log, with the Submarine Signal Company's receiving apparatus, and with wireless telegraphy.

An engineers' workshop, located adjacent to the engine room, is fitted with a lathe, drill press and double carborundum wheels, all power driven.


Upper Bridge Dk
Bridge Dk.
Upper Dk.
Main Dk.
Orlop Deck
Beam on every Frame 5x3x7/20
Beam on Alt. Frames 4x3x7/20
Spaced 26"
17'6"
12'0"
Face Plate 15x1 9/20
Bracket 15x15x7/20
Bridge and Poop Dk Rails as shown
Rail on F'c'le Dk. 3'3" high-2 Rails
Rail Stanchions spaced 4'4" apart
Poop and F'c'le 9/20
Cargo Port
Beam on every Frame F'c'le Bridge and Poop Dks.
9'0" Boiler and Eng. Casing
4¼" dia. Spaced 21-8
8'0" Hatch
Camber 12 in 54'0
1'0" Boiler and Eng. Casing
10" O.D. x 1 9/20 spaced 21'8" apart
Beam on Alt. Frames
Beam on every Frame 7x3.45x3.45x9/20 [Abreast of Hatches and from 23 to 81 Incl.
Hatch End Beams in Way of Bridge and Poop Decks 12x3.42x3.42x15/20 [
9/20 in Way of Cargo Ports
Clips Single
Fore and After at Hatches
14" O.D. x 15/20 spaced 21'8" apart
Orlop Deck Plating 7/20 Oiltight
No Camber
Pillars from Top of D.B. to Main Dk.
17" O.D. x 1½ spaced 21'8" apart
Single Face Bar
Toe same as Frame
Upper Side Stringer
Middle Side Stringer
Lower Side Stringer
Molded Depth 35'4"
Cutting Line of Brackets
Bilge Keel 12x1 1/20 for 200'0" Amidships
Double Bottom
Margin Plate
2nd Long.
1st Long.
27'0" Molded ½ Breadth

MIDSHIP SECTION, SHOWING SCANTLINGS.

PROPELLING MACHINERY.

The propelling machinery consists of one four-cylinder triple-expansion engine and six main Scotch boilers. The cylinders are 35 inches, 58 inches and two of 69 inches diameter, respectively, with a 60-inch stroke, arranged with the high-pressure and medium-pressure cylinders in the center and the low-pressure cylinders at the ends. A separate liner is fitted in the high-pressure cylinder. One piston valve is fitted on the high-pressure, two piston valves on the medium-pressure cylinder, and each low-pressure cylinder has a double-ported balanced slide valve. All valves are worked by the Stephenson double-bar link motion, with direct-acting steam reversing gear; United States metallic packing is fitted to all piston rods and valve stems. All pistons are cast iron of box section, fitted with deep bull-ring and snap rings.

The piston rods are steel, all being 7¾ inches in diameter, and secured to the pistons and cross-heads in the usual manner with taper ends and nuts. The cross-heads are of forged steel with double cast iron slippers having white metal faces.

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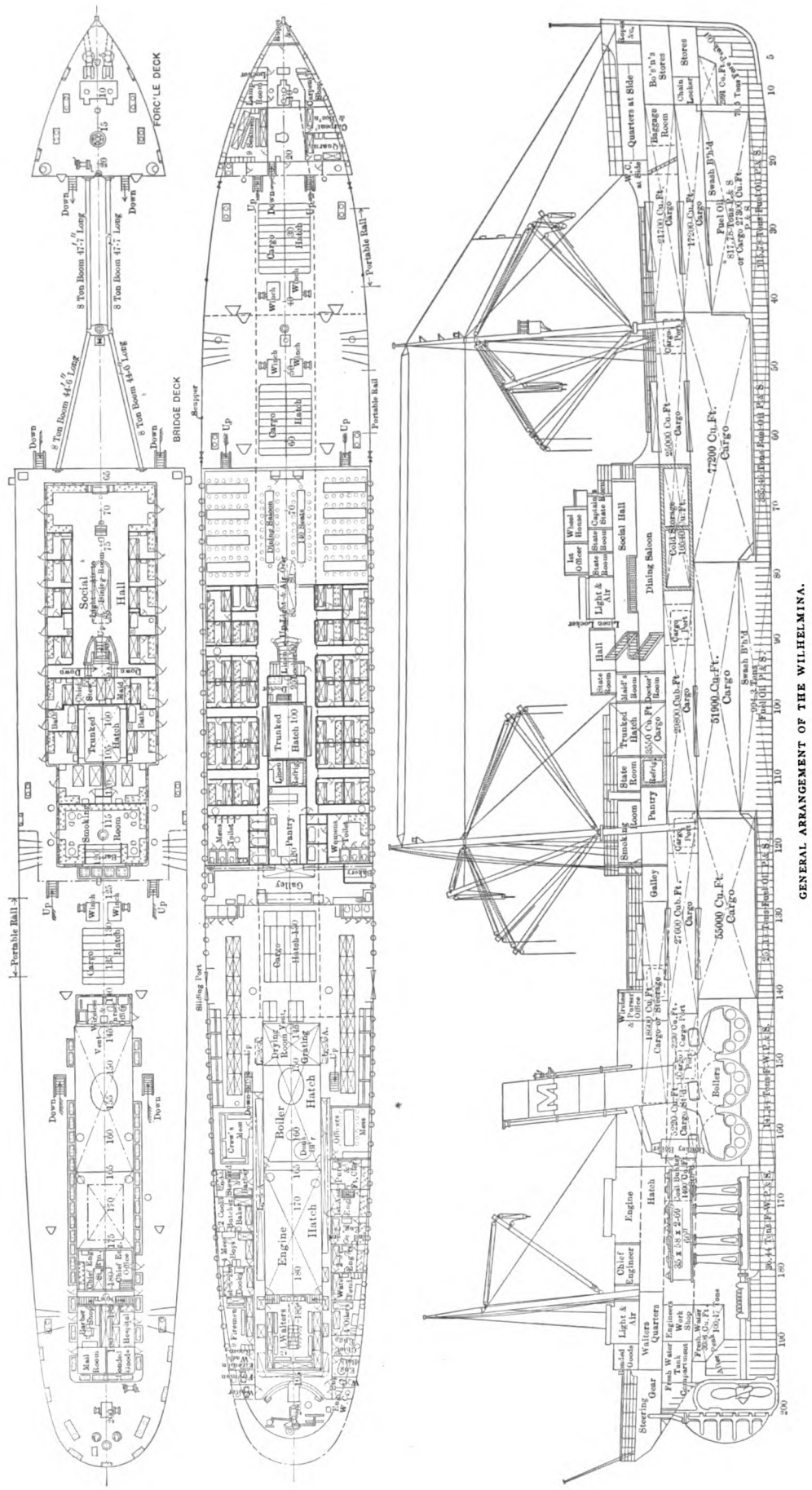

GENERAL ARRANGEMENT OF THE WILHELMINA.

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The connecting rods are forged steel, 11 feet 3 inches between centers, forked at the upper end to suit the cross-head, with gudgeon bearings of composition and crank-end boxes of cast steel lined with white metal. The crankshaft is steel, of the built-up type, with wrought iron webs. It is 19¼ inches diameter, and is made in two interchangeable sections. The sequence of cranks is high-pressure, medium-pressure, forward low-pressure and aft low-pressure.

The bed plate is cast iron of box section, made in three pieces, and has eight main bearings. The main bearing boxes are cast steel lined with white metal, and have wrought iron binders. The housings are also made of cast iron of box section, and have cross-head guides both front and back. The thrust bearing is of the horseshoe type, and the propeller is a four-blade sectional one, with manganese bronze blades. The surface condenser is independent of the main engine, and is made cylindrical with a plate shell. The circulating pump is of the centrifugal type, driven by a single-cylinder, direct-acting engine, and the main air pump is an independent, vertical twin Blake pump. There is also an independent auxilary air pump for port use. All pumps are independent of the main engine, and were furnished by the Blake & Knowles Steam Pump Works. These comprise a vertical twin main air pump, a vertical simplex featherweight auxiliary air pump and donkey feed, fire and bilge, ballast, sanitary, engine-room bilge, fresh water, two evaporator feed and three fuel oil pumps, all horizontal duplex. A Reilly feed heater and Ross grease extractor are fitted; also an evaporating plant consisting of two Reilly evaporators having a capacity of 20 tons in twenty-four hours, and one Reilly distiller of 2,000 gallons capacity in twenty-four hours.

The six main boilers are each 15 feet 4 inches diameter by 12 feet long, built for a working pressure of 190 pounds. Each boiler has four furnaces, 39 inches diameter, with a separate combustion chamber for each furnace and with 364 3-inch tubes. The total grate surface is 486 square feet, and the heating surface 17,070 square feet. Steam drums are fitted on each main boiler.

The donkey boiler is vertical, 5 feet 6 inches diameter by 9 feet 6 inches high, with a submerged head, built for 140 pounds working pressure. The main boilers are arranged to burn fuel oil with steam atomization, and the donkey boiler is arranged for using both coal and oil.

CLASSIFICATION.

The vessel and its equipment, also the propelling machinery, are being built under Lloyd's special survey to Class 100 A1. The vessel will also be rated under the Ocean Mail Subsidy Act of March 3, 1891, for which purpose an auxiliary fire main has been provided below the waterline, and foundations have been fitted for four 6-inch guns.

Lloyd's Register of Shipping—Annual Report, 1908-9.

At the close of the year ended June 30, 1909, 10,424 merchant vessels, registering over twenty and one-half million tons gross, held classes assigned by the committee of *Lloyd's Register*. The details are as follow:

Material of Construction.	Description	British.		Foreign.		Total.	
		No.	Tonnage.	No.	Tonnage.	No.	Tonnage.
Iron and steel	Steam	6,095	12,280,467	2,899	6,390,392	8,994	18,670,859
	Sail	511	836,109	710	1,031,616	1,221	1,867,725
Wood and composite	Steam & sail	197	27,056	12	3,936	209	30,992
Total		6,803	13,143,632	3,621	7,425,944	10,424	20,569,576

The serious depression which has existed for so long a time in the shipbuilding industry is again reflected in the amount of tonnage classed by the society during the year, which is considerably below the very high average attained in recent years.

Classes were assigned by the committee to 550 new vessels. Their registered gross tonnage amounted to 854,984 tons. Of these vessels, 481 of 845,719 tons were steamers, and 69 of 9,265 tons were sailing ships.

Of the total, 470,137 tons, or 55 percent, were built for the United Kingdom, and 384,847 tons, or 45 percent, for the British Colonies and foreign countries.

The most important fact which calls for mention in the work of the society during the past year is the completion of the task of revising the society's rules for the construction of steel ships, which occupied the attention of the committee for several months. The old rules, originally adopted many years ago, had been kept up to date by means of amendments and additions made by the committee from time to time, and proved sufficiently adaptable to be applied as a standard of strength to the various new designs produced by naval architects to meet the requirements of modern oversea trade. Of late years, however, the evolution of cargo-carrying vessels has made rapid progress, and the modification of many types had reached a stage at which it seemed that a general revision of the society's rules was desirable.

Since June, 1908, 44 steamers of upwards of 5,000 tons each have received the 100A1 class, and the following vessels, each of which exceeds 10,000 tons, have been assigned this classification since the issue of the last annual report, viz.:

Name of Vessel.	Tons.	Owners.
Chiyo Maru	13,426	Toyo Kisen Kaisha.
Osterley	12,129	
Otranto	12,124	
Otway	12,077	Orient Steam Nav. Co., Lim.
Orsova	12,036	
Morea	10,890	Peninsular & Oriental S. N. Co
Malwa	10,887	
Mantua	10,895	

It may be mentioned that during the period under review seven steamers, each over 8,000 tons gross, of which the turbine steamship *Chiyo Maru* is the largest, have been built in Japan to class 100A1, and in each case, except that of the above-named vessel, the machinery also has been made in that country.

At the present time, in addition to a fifth liner for the Orient Company, similar to the four previously mentioned, the new Cunard steamship, which is to take the place of the *Slavonia*, is being built to the society's highest class, as are also two new Union-Castle liners, each of 13,000 tons.

Reference may also be made to some other interesting vessels which have recently received, or are being built to, the 100A1 class, such as the two twin-screw steamers for the Canadian Government, the *Earl Grey*, to be used as an ice-breaker and specially fitted up for the passenger and mail service between the mainland and Prince Edward Island, and the *Simcoe*, for lake service, and constructed with special appliances for lifting light buoys; the *Monitoria*, built at Sunderland, representing a novel design of construction in the form of two corrugations worked in the shell plating of the sides, which, it is claimed, will have the effect of diminishing the resistance of the vessel. Further reference may be made to the classification by the society of vessels built on Mr. Isherwood's longitudinal system of construction. Up to the present, thirteen vessels of this type with a total tonnage of 60,000 tons have been, or are intended to be, classed in the register book.

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PRACTICAL EXPERIENCES OF MARINE ENGINEERS.*

Incidents Relating to the Design, Care and Handling of Marine Engines, Boilers and Auxiliaries; Breakdowns at Sea and Repairs as Told by our Readers.

Repairing a Broken Crankshaft at Sea.

The breakdown occurred on board a foreign-owned passenger and cargo vessel in which the writer was serving. The ship was an old one, having been built in the early sixties, with exceptionally heavy scantling. It was driven by compound engines, supplied with steam at a working pressure of 60 pounds per square inch. The shafting was about twenty-five years old, but showed no outward sign of deterioration, and the first indication of anything being wrong was the discovery by the engineer on watch of a fore-and-aft movement on the tunnel shaft, and up to the after main bearing. The forward end of the crankshaft was running perfectly true,

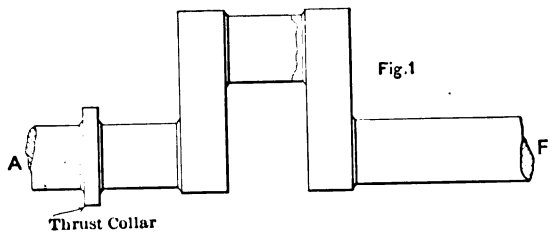

Fig.1

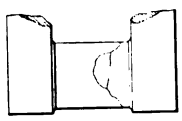
Fig.2

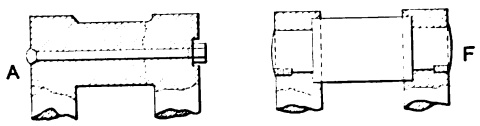

Fig.3 Fig.4

and there was no undue heating of any part of the engines. A light, fair breeze was blowing, with a short and choppy following sea, causing a slight, easy racing of the engines.

Immediately the discovery was made the engines were slowed down and an examination hurriedly made. The inspection showed that the low-pressure crank pin was broken. The engines were then stopped, the captain advised, and the crankshaft dismantled. After dismantling, the fracture of the crank pin was found to be at a point close to the bottom of the fillet, and extending almost half way round, taking afterwards a diagonal course across the pin and out towards the center, as shown in Figs. 1 and 2.

The loose particles of the metal were cleaned off, the parts brought together in position, faired, and firmly secured in place by fitting a band around the pin, the band being drawn firmly together with bolts and nuts. Wedges were driven in between the webs and the engine seating, to enable a hole to be drilled through the webs and the center of the crank pin. The only material available for making a bolt suitable was a tie rod 2 inches in diameter, taken from the deck fittings. Drills were forged from a steel bar to suit this size, and drilling operations commenced. When the hole was drilled through the 32 inches of metal, it was not 1/16 inch out. One end of the hole was countersunk to receive the head of the bolt, and the other end was recessed to receive the nut, as shown in Fig. 3. The bolt with a countersunk head and 1½-inch screw, this being the largest-sized die on board, was then fitted and tightened up without putting any undue strain on it. The crankshaft was then coupled up, the engine started at a slow speed, which was gradually increased up to forty-two revolutions per minute, and at this speed the ship made her port two and a half days later, and only 40 hours late.

It will be observed that the end of the fracture, acting as a clutch, assisted considerably in reducing the shearing strain on the bolt, and also saved a good deal of labor. If the break had been a through one, more than one bolt would have been required, as, with the drilling appliances available, a larger diameter of bolt could not have been fitted.

On arrival in port it was decided to repair the shaft rather than wait for a new one, which would have involved considerable delay. The broken pin was cut away and eye-ways bored out in the webs to receive the new pin. A new pin was made, 12 inches in diameter, the eyes being turned out to this size 2 inches deep, and then reduced to 7 inches in diameter on the forward web, and 8 inches diameter on the after web, with a countersink on each, as shown in Fig. 4. The new pin was carefully finished off and shrunk into place. The ends were riveted over, following up the countersinks, and to prevent any possibility of the pin moving, two 1½-inch tapping holes were drilled, half in the pin and half in the web, tapped, and pins fitted into them and riveted over.

With the crankshaft so repaired the ship made two and a half voyages, covering a distance of about 20,000 miles, and during that time the shaft gave no trouble. It was then replaced by a new one, the repaired shaft being retained as spare. JAMES BELL, R. N. R.

The Emergency Man.

A small government steamer in the service of the National Board of Health was at one time 65 miles from her station on an important mission in connection with yellow fever. This vessel did not require licensed officers, but the rule was to employ them when licensed men could be obtained who cared to take the risk. On this occasion the master of the vessel was only nineteen years old, but he found a way to get his boat back to her station, where repairs could be made, after a licensed engineer had failed completely, giving up the job in disgust and stating that they would have to go somewhere for assistance with a small boat or else wait for some passing craft to pick them up.

The trouble was with the check valves in the feed line, which failed to function, cutting off the supply of water to the boiler. Of course, as soon as the trouble had been discovered, the fires had been drawn and the boat anchored.

The nineteen-year old captain then proceeded to use his ingenuity by making blank flanges in place of the feed-water connection. He then proceeded to fill the boiler with buckets through the safety valve, and after this was done started the fires again. It was found that the boat could make from 15 to 20 miles on every boiler full of water, according to the state of the tide, or on every tack, as a sailor would say. By continuing this performance the youthful captain was able to get his boat back to her station. VERITAS.

* We pay for these articles.

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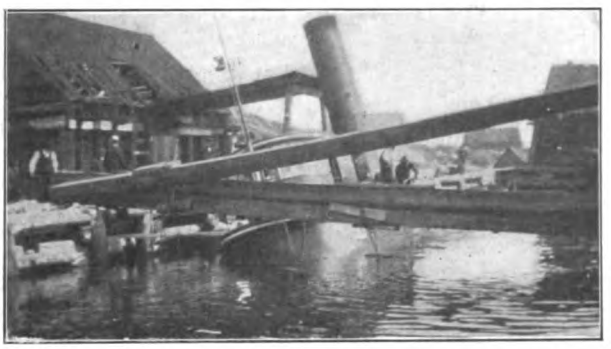
Raising a Sunken Tug.

The photograph shows a tug which sank in about 14 feet of water at her dock. She was raised by placing three steel wire hawsers under the boat and taking the ends to pitch pine timbers, which were placed across and supported one end on the wharf and the other end on a large scow which had been filled with water. Hydraulic jacks were then placed under the ends of the timbers at the wharf and pumped up while the scow was being pumped out. By this means the tug was raised until her rail was level with the water. A



ARRANGEMENT OF TIMBERS FOR RAISING A TUG.

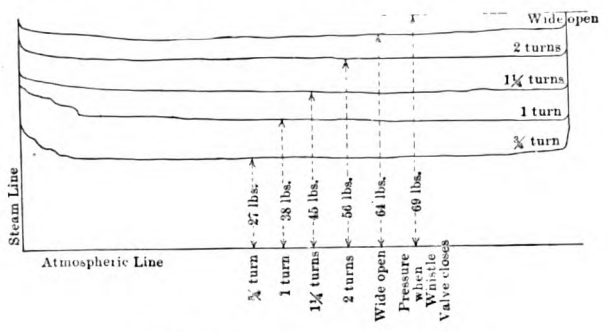
centrifugal pump with an 8-inch suction was then let down through the deck into the cabin and a fire engine and three tug boats pumped the water out of the boat.

At the first attempt the centrifugal pump was stopped by an oil cloth off the mess room table being drawn into the strainer. When this was cleared, however, she was pumped out in about three-quarters of an hour.

Before raising her an attempt was made to pump the boiler out, a pipe being passed down through the dome between the tubes, but since the sea cock on the feed pump had been left open the water flowed in through the checks as fast as the boiler was pumped out. J. F. W.

Steam and Water Test of a 12-Inch Chime Whistle.

The following test was undertaken to show the amount of water used by a 12-inch steam chime whistle according to the pressure in the whistle pipe. The pressure on the gage at the time of the test was 70 pounds per square inch. A 3-inch Williams valve was fitted to the pipe as a stop valve, this being opened at different times three-quarters, one, one and one-quarter and two turns and wide open. Indicator cards were taken showing the drop of pressure in the steam pipe at these different positions of the valve, with the results shown in the illustration. The valve stem was threaded where it passed through the yoke, with threads having a pitch of 6 to the inch. The whistle was allowed to blow 10 seconds each minute.



INDICATION CARDS FROM WHISTLE PIPE.

The computations made to determine the number of gallons of water discharged per hour were as follows:

The velocity of steam in feet per second was multiplied by the length of the blast in seconds per minute, giving the lineal feet of steam per minute; and this multiplied by the area of the pipe gave the number of cubic inches in 1 minute, which, in turn, was reduced to cubic inches per hour. This result was then divided by the cubical proportion of the water discharged, the result, in turn, being reduced from cubic inches to gallons, giving the number of gallons of steam and water discharged per hour.

Valve open one turn, pressure 38 pounds:

$1{,}800 \times 10 = 18{,}000$ feet per minute.
$18{,}000 \times 7.0686 = 127{,}234$ cubic inches per minute.

$$\frac{127{,}234 \times 60}{673.7 \times 231} = 48 \text{ gallons per hour.}$$

Valve open one and one-quarter turns, pressure 45 pounds:

$2{,}000 \times 10 = 20{,}000$ feet per minute.
$20{,}000 \times 7.0686 = 141{,}372$ cubic inches.

$$\frac{141{,}373 \times 60}{563 \times 231} = 65 \text{ gallons per hour.}$$

Valve open two turns, pressure 56 pounds:

$2{,}200 \times 10 = 22{,}000$ feet per minute.
$22{,}000 \times 7.0686 = 155{,}320$ cubic inches per minute.

$$\frac{155{,}320 \times 60}{467.9 \times 231} = 86 \text{ gallons per hour.}$$

Valve wide open, pressure 64 pounds:

$2{,}300 \times 10 = 23{,}000$ feet per minute.
$23{,}000 \times 7.0686 = 162{,}577$ cubic inches per minute.

$$\frac{162{,}577 \times 60}{412.6 \times 231} = 102 \text{ gallons per hour.}$$

INVESTIGATOR.

Main Engine Repairs Under Way.*

On Dec. 13, 1908, the *Kansas* arrived at Colombo, Ceylon, where the fleet was scheduled to remain one week. Such routine overhauling and examination as was possible was begun as soon as the engines were cool; this naturally included the removing the bull's-eyes from the cylinder bonnets, and examining and oiling the interiors. The port engine cylinders were first opened, and everything was found to be in good shape. On the 16th work was begun on the starboard engine, and when the high-pressure cylinder was opened a very remarkable state of affairs was disclosed. There were seen to be two parallel scores, ⅜ inch deep and 1 inch wide, cut neatly in the liner on the forward side, and, in addition, a bad shoulder had been worn on the outboard after portion of the circumference near the top. Removal of the piston follower developed the fact that the packing ring was broken in five pieces, one of which was missing. Although searched for in the other cylinders and valve chests, no traces of the missing piece have ever been found. Probably it has been ground up, or else it remains hidden in some receiver space.

The place of the missing piece had been taken by the forged steel clamp which, as shown in Fig. 1, held the ends of the ring together, making the whole equivalent to a solid ring. This clamp was very securely held in place by the pieces of broken ring and bent and broken springs, and formed a splendid planing tool. The fact that the remaining portion of

* From the Journal of the U. S. Naval Institute.

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the ring was unable to move in the piston was probably responsible for the shoulder.

The damage had been done while en route from Manila, as all cylinders and rings had been examined prior to leaving that port. There was no indication of anything wrong inside the cylinder on this run, except an occasional clicking, to which little attention was paid, as the metallic packing has frequently caused exactly the same sound. The increased coal expenditure was noticed, but was attributed to the stay in Manila Bay, where vessels foul very rapidly. Indeed, on arrival at Colombo, the propellers were examined by divers, and found to be thickly covered with good-sized barnacles. When these were removed it was expected that the coal consumption would resume its former ratio to that of the other vessels of the *Kansas* class. But this was before the damage to the S. H. P. liner was discovered.

There were but three days left of the stay in Colombo, and repairs of any description were out of the question in that time. Remaining behind was just as much out of the question. For almost exactly one year the fleet had remained intact, had made its runs on schedule time, and, judging from newspaper comment, had won the admiration of the world. To confess that one vessel was in such condition as to be unable to proceed would be to admit that a fleet of battleships could not make a cruise around the world. It was, therefore, essential that the *Kansas* leave Colombo on Dec. 20, and that she accompany the fleet during the rest of the cruise. Of course, running under one engine would allow her to proceed, but her maneuvering qualities would be practically nil, and her coal consumption would be so enormous as to necessitate her putting in at Aden. This would again upset the plans of the commander-in-chief, who had reported that the fleet would be at the Canal by Jan. 4.

Had it not been for the bad shoulder in the liner, both engines could have been run without attempting to stop the leakage of steam through the scores, and the coal consumption would probably not have been much greater than on the run from Manila to Colombo. Had the spare ring been installed without removing the shoulder and truing up the liner, it is almost certain that the ring would have broken and caused additional damage. It was clearly necessary to rebore the liner, although unsafe to bore out enough to remove the scores, which, as mentioned, were ⅜ inch deep, while the thickness of the liner wall was 1½ inches, the working pressure being 250 pounds in the high-pressure valve chest. The filling of the scores would save steam, and the removing of the shoulder would allow a new ring to be fitted without incurring risk of breaking it immediately.

The piston and rod were removed, the liner was lifted out of the cylinder and hoisted to the superstructure, and the work of compounding the starboard engine was begun. This consisted in replacing the cylinder bonnet, removing the connecting rod, high-pressure valve, piston and rod, plugging the piston-rod hole, and blocking the steam ports with the valve rings, which were pressed out by improvised spiders. This was to prevent the high-pressure cylinder from acting as a condenser by presenting a large cooling surface to the incoming steam. The crank pin was carefully wrapped with burlap to keep it from being damaged. The cross-head was secured in place on its guide.

The liner could have been swung in the large gap lathe in the ship's workshop, but, unfortunately, the shop door was too narrow, and cutting away the bulkhead would consume considerable time. The commander-in-chief had ordered the *Panther* to afford all possible aid, and accordingly a boring bar and motor were borrowed from her.

On the morning of Dec. 20 the *Kansas* left Colombo with the fleet, and took her regular place in column. She took part in all tactical exercises, which were held daily, and no difficulty was experienced in handling the starboard engine. The coal consumption was increased to some 15 percent above normal, and approximated very closely to the consumption on the run from Manila, when the engine had been run with the broken ring.

The actual repair work was begun as soon as the vessel cleared the breakwater. To carry out the plans it was necessary to bore out the shoulder, fill in the large scores, enlarge the piston and follower, and make a new ring, as the spare would be useless after reboring. Various materials suggested themselves as suitable for filling in the scores. It was finally decided to fit cast iron strips, as the Babbitt metal on board might run, and "Smooth-On" would probably crack off and ruin all the other cylinder liners. To fit these strips, the sides and bottoms of the scores had to be exactly true, and, while the clamp in its capacity of planing tool had done a very good job, the scores were not absolutely straight. The milling out

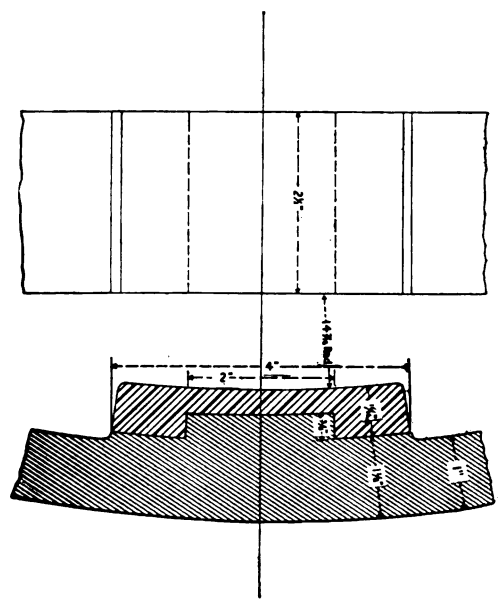

FIG. 1.

of the scores was successfully done by an ingenious device invented by Machinist E. G. Affleck, U. S. N., who had complete charge of the repairs, and who, by working day and night, pushed them to a speedy and successful termination.

The liner was laid in a horizontal position on the superstructure near the engine-room hatch, and that part of the ship was roped off so that the mechanics could work without interruption. A wooden dam was built to keep the water away from the liner and motor while the decks were being washed.

Splendid weather and a smooth sea favored the undertaking. In addition, the Red Sea did not live up to its reputation and the men did not suffer from the heat.

After the scores had been milled, the cast iron strips, made from the furnace-door lintels, were accurately fitted, and driven in place. They were then drilled, and ⅜-inch studs were tapped into both the strips and the liner. It is unfortunate that the strips could not have been each in a single piece, but the material available on board demanded that each score be filled by inserting three strips end to end. As will be seen later, the strips were not entirely successful, and were removed.

The scores being filled, the boring bar was rigged, and three cuts were taken, each cut requiring about ten hours. The liner, after boring, was 33 inches in diameter, whereas the original design called for 32½ inches. The tool was set by assuming the counterbore to be concentric with the liner, and this assumption proved to be justifiable, for careful calipering

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showed that the liner had been bored exactly true, and, when afterward set in place in the cylinder, was perfectly in line.

While the work of boring was going on the piston and rod had been centered in the large lathe, and a cut taken from the piston and follower. Fig. 2 shows the original arrangement, while Fig. 3 illustrates the conditions after the enlargement. The intention was to shrink solid cast iron rings on both piston and follower, and then to turn the whole to suit the new diameter of the liner. A casting of sufficient size to make both enlarging rings, as well as a new packing ring, had been obtained from stock on the *Panther* before leaving Colombo. The shrinking-on was successfully done, but it was while finishing the enlarging rings that a very serious setback was received. One cut was sufficient to disclose the fact that the

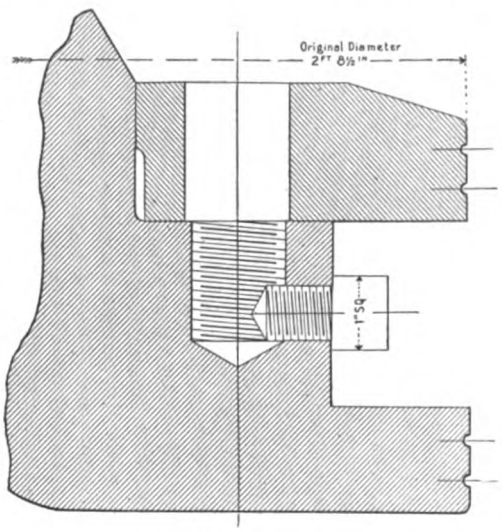

FIG. 2.

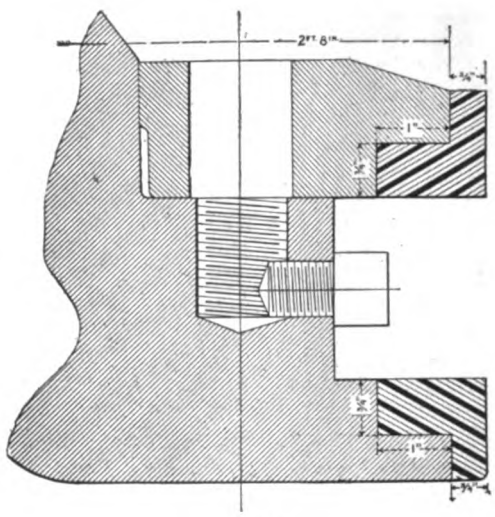

FIG. 3.

casting was so badly honeycombed as to render it useless. Fortunately, the packing ring, made from the same casting, turned out to be sound. It was now necessary to resort to some other expedient, and time was getting short, since the possibility of bad metal had not been figured into the calculations. The engineer's blacksmith took the only piece of 2-inch square steel on board, bent it to a perfect circle of the proper diameter, and welded the ends together. As blacksmithing has to be done in the fire-rooms, where the foundations for the anvil are none too solid, and as all work has to be done by hand, this piece of work may well be considered an exceptional job of ship's blacksmithing. These two blacksmiths, T. Lentz and J. Mullins, deserved the received special commendation for this exhibition of skill.

Two such rings were needed; the second one was ordered by wireless from the *Panther,* and put aboard the *Kansas* on arrival at Suez. Both rings were shrunk on by the engineer's blacksmiths, and the work of turning them up was begun when the ship started through the Canal.

The fleet arrived at Suez on Jan. 3. The first division, comprising the *Connecticut, Vermont, Minnesota* and *Kansas,* proceeded through the Canal on the morning of Jan. 4, and tied up at Port Said about 1 A. M. Jan. 5. Coaling was begun at once, and while in progress hurry orders were received to get out at daylight, Jan. 6, in order to reach the vicinity of the Messina disaster as soon as possible. This left about twenty-eight hours in which to complete all repairs to the starboard engine, and it was not possible to begin work on the engine itself for four hours, on account of the heat of the metal.

Turning the ring received from the *Panther* took a great deal more time than it should have, as it was made of the hardest kind of steel. Taking one light cut made regrinding the Armstrong tools necessary, and these were made of the highest grade of self-hardening, high-speed steel.

When the engines were cool, the liner was secured in the cylinder, the joint under the flange was made, the connecting rod was swung back into place, the cross-head and crank-pin brasses were adjusted, the compounding devices were removed, the piston and rod were put in place, the high-pressure valve was installed and set, the new ring was accurately fitted, and the ship got under way at 14-knot speed at 5 A. M., Jan. 6, headed for Naples.

The repairs thus effected were successful, except that on arrival at Villefranche (to which port the ship had been diverted while passing through the Straits of Messina) one cast iron strip was missing, while another was found to be cracked. Accordingly, the remaining strips were removed, the scores were dovetailed, and white metal from a spare set of United States metallic packing was poured in hot, and peened and scraped in place. On arrival at the Navy Yard, Philadelphia, the liner was in splendid condition, and the places where the scores had been filled in could scarcely be seen.

It is calculated that the rate of coal consumption was practically the same coming from Manila to Colombo, with the ring broken, as it was from Colombo to the Canal with one engine compounded. On the runs from Villefranche to Gibraltar, and thence home, the consumption was decreased 15 percent, resulting in a saving of 230 tons in 4,600 miles.

LIEUT. E. C. KALBFUS, U. S. N.

The Bureau of Navigation of the Department of Commerce and Labor, Washington, D. C., reports that ninety-three ships, both sail and steam craft, were built and officially numbered in October. Of this number more than one-half were constructed for service on the Atlantic and Gulf coasts. Only six were steel steamers. The vessels were small, their total being only a little over 4,000 gross tons.

Recent launches of naval vessels in Britain include the battleship *Neptune,* the battle cruiser *Indefatigible,* and the second-class cruisers *Gloucester* and *Liverpool.* The *Neptune* is an advanced type of Dreadnought, capable of firing all of her ten 12-inch guns on either broadside. The *Indefatigible* will be the most powerful cruiser in the world, with a speed of 26 knots. The two second-class cruisers are of the Bristol class, having a displacement of 5,000 tons and a speed of 25 knots.

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SEVENTEENTH ANNUAL MEETING OF THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS.

The seventeenth annual meeting of the Society of Naval Architects and Marine Engineers was held in the Engineering Societies building, New York, Nov. 18 and 19, 1909. During the year a special effort was made to increase the membership, and, according to the report of the secretary-treasurer, seventy-five new members were elected during the year. There was a membership of 765 Nov. 1, 1908, whereas the figures for Nov. 1, 1909, were 795. Forty-seven new members were elected at the present meeting. The total receipts for the year were $10,063.23 (£2,066), and the total disbursements, $12,785.08 (£2,625). The total resources of the society at present are $24,575.21 (£5,046), with no liabilities against it.

PAPERS READ NOVEMBER 18.

No. 1—Evolution of Screw Propulsion in the United States.

BY CHARLES H. CRAMP.

ABSTRACT OF PART I.

Between the first and second quarters of the last century, the principal machine shops of Great Britain were in a prosperous condition, abundantly supplied with good tools and numerous skilled workmen; while the machine shops in the United States were few in number, limited in size and capacity, and the workmen, with but few exceptions, lacked experience and skill. The relative conditions between the machine works of the two countries remained the same until the year 1870. It was on this account that the screw-propeller, invented by Col. John Stevens in 1804, was not utilized.

The screw of Colonel Stevens of 1804 was practically developed in Kensington by the Penn Works of Reaney, Neafie & Company half a century later, and the hull of the *New Jersey*, which was designed by the best ability in England, was adopted by William Cramp in the screw tug *Sampson*, which was engined by the Penn Works, the propeller of which boat was practically that of Colonel Stevens.

Screw propulsion had already become a factor in navigation, and there was a general desire to develop it, particularly in Philadelphia, during the decade between 1834-1844. All were in a receptive mood, the atmosphere was highly charged with it, and was ready for the great transition when John Ericsson made his appearance in 1840, preceded by the *Robert F. Stockton*, which came over from England under sail the preceding year.

After Mr. Ericsson's arrival here in 1840 he made the acquaintance of Mr. Thomas Clyde, who soon became a convert to the propriety of screw propulsion, and he was the first shipowner to adopt it in this country. A pair of screws was fitted to the *J. S. McKim*, which he used in his Gulf trade, and afterwards as a transport in the Mexican War. The engines were built in a shop in the Old Northern Liberties, of Philadelphia, and Mr. Neafie, afterwards of Reaney, Neafie & Company, was engaged on them as a young man, and it was to this fact that Mr. Neafie took early interest in screw propulsion. Later on, when the firm of Reaney, Neafie & Company was established in 1840, screw propulsion had come to stay, and was the practical ending of the Ericsson screw.

While Mr. Ericsson had accomplished so much in the education of the public mind in favor of screw propulsion, he was not so successful in the introduction of his particular screw; it proved exceedingly defective, and none were used the second time.

The original screw of Colonel Stevens was replacing all others, but not under his name. The favorite screw of Reaney, Neafie & Company was the "Loper patent." Captain Loper had bought the patent from a workingman, and I think he died without knowing anything of its resemblance to the Stevens propeller. Mr. Clyde had adopted it, and had transferred his work to Reaney, Neafie & Company. The Ericsson Line adopted a screw of their own which was much similar to the Stevens.

Thomas Murphy, the chief engineer of the Ericsson Line, was a highly competent and superior man, and he adopted a plan to simplify the construction and repair of "his wheels"; the repairs being frequent made it very costly. Besides, he took advantage of every change he made in their form to secure information as to the results, subjects that still continue to puzzle all marine engineers—the proper relationship between the pitch, diameter, the surface area and the revolutions. He secured a vast fund of information from his multitude of experiments. I do not know whether he left any record with the company after his time, but I do know that the Penn Works profited vastly by his experiments.

The particular object of Mr. Ericsson's visit here was more on account of the introduction of a certain type of engine for warships than of his peculiar screw. He had interested Commodore Stockton in the warship question at an early date, when he met him in England. The Commodore prevailed on the United States Government to build a ship which was named the *Princeton*, and Mr. Ericsson was occupied from September, 1841, to September, 1843, in preparing the plans of the machinery. It is impossible to find in the world's history of naval construction where there had been so great a flourish of trumpets over the building of a ship, with such little results afterwards as in the case of this ship.

Great Britain had at an early date begun to develop screw propulsion, and was building ships and engines of the best character, and introducing them in their ocean commerce and in their navy, which resulted in their paramount influence in ocean commerce of that country. It practically began with the advent of the fine screw ship, the *Great Britain*, in 1844.

While Great Britain was engaged in the establishment of its future pre-eminence, the great commercial city of New York adhered to the paddle-wheel, with its walking-beam engine, that led to our ultimate elimination from the world's commerce.

While the decline of our foreign commerce incident to the prolonged use of the paddle-wheel became a conspicuous feature in the East, it did not destroy our coastwise business, which extended from the East to the Pacific Coast. The business with the cities there, New York and San Francisco was still in the hands of shipping companies of New York, and many small paddle-wheel steamers were still built, but much of the local trade and travel in the Pacific was done in small iron vessels, many of which were screws and built on the Delaware. When the large paddle-wheelers were worn out or lost, they were not renewed. The propellers were generally of Captain Loper's patent, and his name became as much talked of as Ericsson's was previously, particularly as the propeller was much better.

No. 2—The Effect of Parallel Middle Body Upon Resistance.

BY D. W. TAYLOR, NAVAL CONSTRUCTOR, U. S. N.

ABSTRACT.

A portion of the work of the United States model basin during the past year consisted of an investigation into the effect upon resistance of full vessels of varying percentages of parallel middle body. This question arises at times in dealing with full vessels of moderate or low speed, and it appeared desirable to determine whether there was, from the point of view of resistance, an optimum length of parallel middle body in a given case. There are, of course, a number of factors

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entering into the matter in the case of a design, but resistance alone is considered in this paper.

The lines of the parent form from which all the models tested in this connection were derived represent a model having a 'midship-section coefficient of .96, a ratio of beam to draft of 2.5, and a longitudinal (prismatic) coefficient of .68. All the models tested had the same 'midship-section coefficient and ratio of beam to draft, variations being in longitudinal coefficient, in size and in shape of curves of sectional area.

There were three series of models tested, each series containing twenty models. Four sizes of models were used in each series, and for each size of model five curves of sectional area were used. The longitudinal coefficients used were .68, .74 and .80. For the .68 longitudinal coefficients the five percentages of parallel middle body were 0, 9, 18, 27 and 36. For the longitudinal coefficients of .74 these percentages were 0, 12, 24, 36 and 48. For the longitudinal coefficients of .80 the percentages were 0, 15, 30, 45 and 60. The displacements used for the 20-foot models in fresh water were 1,000 pounds, 1,500 pounds, 2,250 pounds, and 3,000 pounds. This range covers the majority of vessels.

Displacement remaining unchanged, the greater the percentage of parallel middle body the finer the ends.

Resistance being due partly to the friction of wetted surface and partly to wave making, etc., constituting the residuary resistance, it is naturally considered under these two heads.

The larger models have a little more wetted surface, as would naturally be expected, owing to the greater obliquity of their waterlines. The difference, however, is slight. The variation with percent of parallel middle body is also very slight.

Practicable variations in the length of parallel middle body in a given case will have hardly any effect upon wetted surface or skin resistance. It should be remembered in this connection that skin resistance is always the major factor of the total resistance for vessels of the kind now under consideration. As a general thing for such vessels when well designed the skin resistance is in the neighborhood of 70 percent of the total.

Considering now the residuary resistance the conditions are very different, and we find this element of the resistance materially affected by the percentage of parallel middle body.

At the very high speeds any parallel middle body is prejudicial to speed. While vessels of full type are not pushed to high speeds, it may be remarked that the high residuary resistance associated with the fine ends at the high speeds is in agreement with a number of other experiments made at the model basin, which indicate that for very high speeds full ends are favorable to speed rather than the reverse. For moderate speeds, however, below a speed-length ratio of unity, full ends are usually distinctly prejudicial to speed.

For practicable speeds for full ships, say below a speed-length ratio of .9, the experimental results indicate an optimum length of parallel middle body, the greatest residuary resistance corresponds to 48 percent of parallel middle body, but the next corresponds to 0 percent, the minimum residuary resistance evidently corresponding to some length of parallel middle body between 0 and 48.

From a series of diagrams, showing contours of residuary resistance in pounds per ton, plotted upon percentages of parallel middle body as abscissæ, and values of displacement-length coefficient as ordinates, it is shown that for a speed-length coefficient of .55, the minimum residuary resistance is almost independent of the displacement-length coefficient, corresponding very closely to 25 percent length of parallel middle body regardless of the displacement-length coefficient. For higher speeds, there is more variation in the value of minimum residuary resistance as we vary displacement-length coefficient, but the variation in length of parallel middle body corresponding to minimum residuary resistance is not great, so we can very reasonably consider the actual minimum residuary resistance between displacement-length coefficients of 100 and 150 as practically constant. Hence we are enabled to almost eliminate the effect of displacement-length coefficient.

Curves plotted upon values of speed-length coefficient showing (1) the minimum residuary resistance; (2) the percentage of parallel middle body corresponding to the minimum resistance; (3) the resistance 10 percent above the minimum; (4) percentages of parallel middle body above and below the minimum for which the residuary resistance is 10 percent above the minimum, indicate the boundaries within which length of parallel middle body may be varied without appreciable effect upon resistance. Thus, if the residuary resistance is about 30 percent of the whole, it may be increased 10 percent with an increase of but 3 percent in the whole resistance. The upper limit is naturally the more important, and the experiments indicate that if the lengths of parallel middle body given by the curves fixing the upper limits are much exceeded, there is material increase of residuary resistance.

Broadly speaking, from the point of view of resistance alone, for the range of speeds attained in practice by full vessels, the optimum length of parallel middle body is for longitudinal coefficient of .68 from 12 to 16 percent, but it may be made 25 percent without material increase in resistance. For a longitudinal coefficient of .74 the optimum length of parallel middle body is from 24 to 27 percent, but it may be made from 36 to 40 percent without material increase of resistance. For a longitudinal coefficient of .80 the optimum length of parallel middle body is from 32 to 35 percent, but it may be made from 44 to 48 percent without material increase of resistance.

These conclusions apply to values of speed-length coefficient above .50. For very low-speed vessels the residuary resistance is such a small percentage of the total that the limits above may evidently be materially exceeded.

No. 3.—The Influence of the Position of the Midship Section upon the Resistance of Some Forms of Vessels.

BY PROFESSOR H. C. SADLER.

ARSTRACT.

In the following experiments the term "midship section" designates the section of maximum area. Two sets of models of ordinary form were tried, and in each series the length, breadth, draft, displacement, sections and curve of sectional areas were kept constant, the only variation in the form being that due to expanding or contracting the forward and after body, due to placing the midship section at various positions in the length. When the midship section is at the center of the length, the curves of sectional areas in both models were the same for both forward and after bodies:

PARTICULARS OF MODELS.

Model.	$\frac{L}{B}$	$\frac{B}{d}$	$\frac{L}{d}$	Coefficients.		
				Block.	Prism.	Mid. Sect.
A	8	2.143	17.14	.503	.538	.935
B	8	2.143	17.14	.567	.606	.936

In each case the midship section was placed in four positions: (1) At the center of the length; (2) at 5 percent of the length aft of the center; (3) at 10 percent aft, and (4) at 10 percent forward of the center.

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At low speeds the position of the midship section has little or no influence upon the resistance, but as the speed increases there is a certain position for each speed where the resistance is a minimum. This position travels aft as the speed increases. With the midship section 10 percent aft or 10 percent forward of the middle of the length, the resistance shows a marked increase at the higher speeds.

The curves for model B follow in the same general lines as those for model A. In this case, however, as the model is of somewhat fuller form, the minimum occurs at a slightly different place, and the effect of the position of the midship section is noticeable at a smaller speed-length ratio.

In both models, when the midship section was placed at 10 percent of the length aft of the center, the flow around the after body seemed somewhat disturbed. This was doubtless due to the hollow form caused by the closing up of the sections, and the performance in this particular case could doubtless be improved by filling out the after body at the stern and easing the form immediately aft of the midship section. Such modifications, however, would destroy the main object of the above investigations, which show the effect of the position of the midship section only, everything else remaining constant.

No. 4.—Some Ship-shaped Stream Forms.

BY ASSISTANT NAVAL CONSTRUCTOR WILLIAM M'ENTEE, U. S. N.

ABSTRACT

The investigation on which these notes are based had for its principal object the determination of the variation of velocity of water when constrained to move in a plane along a form of the shape of a ship's waterline, or, more properly, waterplane.

The main conclusions from the investigation are: (1) That hollow waterlines cause less wave-making disturbance than straight or convex waterlines, and (2) that, as the hollowness and fineness of waterlines are increased, the wave-making disturbance decreases to a minimum, after which, if the lines are made still finer and hollower, the wave-making disturbance again increases.

These conclusions result from an investigation which, to extend to actual ships, requires assumptions which seem reasonable, but for which no rigorous proof is given.

It was early seen that previous investigations of the subject dealt with stream forms, which had points of variance from those which appeared to be of most interest. The variation was due to using sink and source functions of such form that the resulting stream forms were (1) blunt or rounded at the entrance point, (2) the entrance was convex and neither straight nor hollow. The principal difficulty with such forms is that, no matter how fine they are made, the pressure at the entrance point is always that due to the velocity-head of the speed of advance, a condition which prevents comparison of the probable wave-making effect of waterline of varying sharpness of entrance and varying waterline coefficients.

Since waves about a ship are due to variations in pressure head in the water around it, it seems obvious that the greater such variations the greater will be the wave disturbance, and as the pressure head depends on the velocity, it appears that the conclusion to be drawn from the above comparison, in which the form which has the hollowest lines and smallest waterline coefficient has the greatest velocity variation along it, is that the waterline coefficient may be made too small and the waterline too fine and hollow. The bow wave may be smaller, but the depression amidships is greater and the total wave-making result may also be greater.

Probably the most interesting conclusion from the investigation is the desirability of using hollow waterlines to reduce wave-making. It can be shown that if a stream form has a straight or convex line at the entrance, the pressure head will be a maximum and equal to that due to the velocity through the water. The total variation of velocity along such a stream form will be more than 100 percent.

No. 5—Applications of Electricity to Propulsion of Naval Vessels.

BY W. L. R. EMMET.

(This paper is published in full on page 469.)

No. 6.—The Producer-Gas Boat Marenging.

BY H. L. ALDRICH.

ABSTRACT.

This paper describes in detail the arrangement and performance of the 40-foot cruising motor boat *Marenging*, equipped with a heavy-duty, four-cylinder, four-cycle engine operating on producer gas. The substance of the paper will be found on pages 110 and 111 of the March, 1909, and page 313 of the August, 1909, issues of INTERNATIONAL MARINE ENGINEERING. As the result of his experience with this boat, Mr. Aldrich states that, as compared with gasoline (petrol), it costs scarcely one-tenth as much to use producer gas, and as compared with steam, probably from one-third to one-half as much.

No. 7—Building and Equipping the Non-Magnetic Auxiliary Yacht Carnegie with Producer-Gas Propelling Equipment.

BY WALLACE DOWNEY.

ABSTRACT.

This paper describes briefly the construction of the hull and machinery of the 568-ton auxiliary yacht *Carnegie*, built entirely of non-magnetic materials for the Carnegie Institution in Washington, to be used in making a survey of the magnetic variations at sea. The yacht has a brigantine rig, and is specially fitted for taking magnetic observations. The auxiliary propelling machinery consists of a 150-horsepower gas producer and engine, built entirely of non-magnetic materials. A complete description of this vessel, covering the substance of this paper, was published on page 47 of the February, 1909, issue of INTERNATIONAL MARINE ENGINEERING.

PAPERS READ NOVEMBER 19.

No. 8—The Design of Submarines.

BY MARLEY F. HAY.

ABSTRACT.

All submarines are fundamentally alike, the greater part of the given reserve buoyancy being neutralized by the addition of water to a main tank, an auxiliary tank being provided to compensate for different weights and the difference of salinity in the sea water. Beyond this point, however, submarines differ according to the judgment of the designer, the main factor which seems to be responsible for all the divergence of ideas being the safety factor.

Submarines may be divided into two classes, the double-hull and single-hull. In double-hull submarines, the outer hull is designed on torpedo boat lines, in order to develop a high surface speed. As this shape of hull is incapable of sustaining the pressure due to great depth, an inner hull is fitted for strength. In boats of this type the depth to which the boat may safely go is less than in the case of single-hull vessels, the maximum for a Krupp or Laurenti boat being 43 meters, while a single-hull boat of the Holland type has been submerged to 70 meters, and is capable of submergence to 100 meters. On account of

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the additional weight of the double-hull vessel its factor of safety in a vertical plane is low, and an attempt to increase this factor is usually made by fitting a drop keel. The necessity for a safety keel is not so urgent in a single-hull vessel, and water ballast which can be blown out by compressed air is substituted.

Coming to the question of stability and seaworthiness in the double-hull vessel, the center of gravity of hull and machinery weights is above the center of buoyancy when the ship is at the surface. Therefore, when being submerged the center of buoyancy rises and the center of gravity drops, there being a critical point where they coincide, and the stability of the vessel must depend entirely on its form. In single-hull vessels the center of gravity is always below the center of buoyancy. Due to the considerable metacentric height of the double-hull vessels, the rolling period is very short, and the angle of heel is liable to become excessive, whereas in the single-hull vessel the rolling period is lengthened, stiffness is much decreased, and, in fact, the rolling is practically replaced by a curious lateral translation of the entire ship.

Double-hull vessels can be constructed with large percentages of reserve buoyancy by making the superstructure watertight.

It was formerly considered that submergence could be carried out in two ways. First, by submerging on an even keel by means of double rudders, or hydroplanes, forward and aft, and, second, by inclining the axis of the boat downwards by means of a single diving rudder at the stern. As a matter of fact, the hydroplane system acts in the same manner as the single diving rudder, and this difference really no longer exists.

Although double-hull vessels show high surface speed, yet the weight of the accumulator battery is so cut down that the submerged speed is relatively low and the radius of action when submerged comparatively short. Single-hull vessels are capable of higher submerged speeds. Which of these qualities is desirable, however, depends upon tactical considerations. As a rule, single-hull vessels maneuver more easily.

The tendency of future submarine design will be towards vessels of higher surface speed for which a single-hull vessel can undoubtedly be designed, the adoption of heavy oil engines to increase the safety of fuel storage and, possibly, the development of an internal-combustion engine, running on a closed cycle, which can be used when the boat is submerged, therefore, eliminating the accumulator battery weights.

No. 9—The Foreign Trade Merchant Marine of the United States. Can it be Revived?

BY G. W. DICKIE.

ABSTRACT.

A careful study of the early history of the foreign trade shipping of the United States will show that the foreign shipping trade of this country never prospered even in the days of wooden ships without substantial protection, and this is due to the fact that every nation which aspires to maritime power has been willing to pay for it.

In 1789, when protection began, the foreign trade shipping of the United States amounted to only 123,893 tons, including both exports and imports carried in American bottoms, and forming but 17½ percent of the imports and 30 percent of the exports. In ten years this had increased to 657,142 tons, forming 90 percent of the imports and 87 percent of the exports. During the war of 1812 the figures fell to 71 percent of the imports and 51 percent of the exports! Under the original navigation laws, in 1825 the United States carried in her own ships 95.2 percent of her imports and 89.6 percent of her exports, and this is the highest point reached by American shipping in foreign trade.

Many think that the navigation laws which were formerly so effective would again give to the United States her fair share in foreign trade. While this is doubtless true, there is grave doubt as to the possibility of maintaining under such laws pleasant relationship with other nations who are doing business on the ocean.

It is unnecessary to describe the existing conditions as far as the merchant marine in foreign trade is concerned. About 93 percent of our foreign commerce is being carried in foreign ships under foreign flags. Contrary to the general impression, the decline of our foreign shipping trade during and since the Civil War was not due primarily to the war; on the contrary, the steady decline began from the time (1812) that the foreign-carrying trade of the country was opened free to foreign ships, when the final restrictions on competition by all countries in our foreign trade were removed.

In addition to the laws of protection to American shipping in the foreign trade, other forces have hastened its decline. One of these was the change from wood to iron, and then steel, in the construction of ships, and the substitution of steamships for sailing vessels.

When this country once realizes that the time has come when it is a national necessity that merchant ships built in our own shipyards, officered and, if possible, manned by our own citizens, owned and operated by our progressive men of affairs, shall represent our enterprise and power in all parts of the world, there will be found a way to do it with profit to all concerned. Efforts to stimulate shipping will then be understood by the people, and questions regarding such matters and needing legislation will be treated in the manner that their importance demands. Admitting to register foreign-built ships, as now proposed in a measure before Congress, will not revive the shipping of this country; if it would the shipbuilder might be willing to be sacrificed in order that such a result might follow. A country that could not build ships has never, as far as I have been able to find, been able to own and operate them.

The necessity of doing our own foreign-carrying trade is fast growing. Under present conditions the value of our exports in round figures is $1,750,000,000 (£360,000,000), and of our imports, $1,250,000,000 (£256,600,000). About $250,000,000 (£51,300,000) are paid annually to foreign ship owners to carry on this commerce.

There are reasons why the American ship cannot compete with the foreign ship in the ocean-carrying trade. National conditions over which our shipbuilders or shipowners have no control, and which they are powerless to change, make the cost of building vessels in the United States from 30 to 40 percent greater than in other countries. The cost of manning and victualing these American ships is also much greater, probably not less than 30 percent more than manning and victualing foreign ships. In addition there are other expenses in the operation of vessels which are greater in the United States than they are in other countries, such as taxes, repairs outfit and equipment. Most of these higher costs are the outgrowth of conditions resulting from the policy of high protection to industries that have been developed under laws first enacted, strange as it may seem, by the very Congress that removed all protection from shipping engaged in the foreign trade, and which policy has continued through all the period that American shipping engaged in the foreign trade has been declining.

Other nations are paying approximately the following amounts for the stimulation of their foreign trade: Great Britain, including Admiralty subventions, $7,000,000 (£1,440,000); France, including bounties on construction and

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UPBUILDING THE AMERICAN MERCHANT MARINE.

To-day the American merchant marine engaged in foreign trade is practically extinct. We have but five steamships regularly crossing the Atlantic to Europe and only six regularly crossing the Pacific. We have no steamships under the American flag on routes to South America below the Isthmus and the Caribbean Sea, and none to Australia or to Africa. The tonnage of American ships engaged in foreign trade is to-day less than it was one hundred years ago. In 1810 approximately 90 percent of our exports and imports were carried in American ships, whereas to-day we are carrying less than 10 percent, and how much more the carrying of this trade means to-day than it did a century ago every one who is conversant with the great industrial and commercial growth of this country knows. It is sufficient to note that between $200,000,000 and $300,000,000 are paid annually to foreign steamship companies for this service.

THE CAUSE FOR THE DECLINE OF AMERICAN SHIPPPING.

The reason for the decline of American shipping in foreign trade is perfectly obvious. There is nothing mysterious or inexplicable about it. It is the natural result of lack of protection by the Federal government. The industry of carrying goods on the high seas is the one unprotected American industry.

Due to our system of protective tariffs, which has affected the cost of nearly everything that enters into the construction of a ship, including shipyard tools and machinery, workmen's wages, and, until recently, most of the materials, the total cost of an American-built ship anywhere from 50 to 100 percent greater than that of a foreign-built ship, precluding the thought of free competition in the foreign carrying trade.

When it comes to the operation of the ships, it is found that in accordance with our navigation laws American seamen must be paid wages ranging from 50 to 100 percent higher than those for foreign seamen, and that their standard of living must be such that the cost is approximately double that for a foreign ship.

Due, then, to high cost of construction and operation, the American ship without government aid of any kind has naturally disappeared from the high seas.

This condition of affairs is now becoming generally recognized throughout the country as one which is detrimental to the best interests of the nation, and one which should be rectified if there is a possible way to do it. In spite of the growing sentiment in favor of upbuilding the American merchant marine there are, however, some who still consider that this step is entirely unnecessary.

The up-building of the American merchant marine is justified by two very strong and convincing reasons. The first is that it would aid the commercial growth of the country by bringing merchants and manufacturers into closer touch with foreign markets, enabling them to take a progressive stand in export trade; and the second is that it is necessary to provide an adequate reserve force of auxiliary ships and men for our navy.

THE MERCHANT MARINE AS AN AID TO COMMERCE.

As to the first reason, it is true the exports and imports of this country are now enormous, and increasing rapidly each year, in spite of the fact that we are dependent upon foreign steamship lines for the carrying of about 90 percent of our goods. It is also true that this work is being done cheaply; but here any advantage in the present system ends. Carrying our commerce in American-built ships, officered and manned by American citizens, and sailed under the American flag, would not only widen old markets but would enable Americans to open up new markets for themselves instead of being obliged to follow the lead of their competitors.

The foreign trade of South America is increasing at the rate of $100,000,000 a year. China is awakening, and a few years will undoubtedly bring about an immense growth in her foreign trade. Japan's commercial activity is well known. In all this growth of trade and commerce the United States, on account of her vast resources and great industrial enterprise, has an opportunity for acquiring power, greatness and prosperity beyond the possibilities of any other nation. To expect to accomplish this, however, while dependent upon foreigners for the transportation and delivery of our products, is as absurd as to expect one merchant to out-distance a rival to whom he entrusts the important task of delivering his goods. This is precisely what we are doing at the present time, and the overwhelming disadvantages of such a policy are clearly apparent in the difficulties which our merchants and manufacturers encounter in entering the South American trade and staying there in competition with the direct and efficient mail and freight steamship lines which other nations maintain to that country. Every ship is a missionary of trade, and steamship lines mean as much to the development of the countries to which they belong as railroads do to their terminals.

IMPORTANCE OF THE MERCHANT MARINE TO THE NAVY.

The second important reason for upbuilding our merchant marine is the urgent necessity of providing an adequate supply of auxiliary ships for our large and otherwise well-equipped navy, and of forming a strong naval reserve from which recruits can be drawn for our warships in time of war. The present lack of such resource places us in a position that is at once humiliating and serious. Only recently we have had this fact impressed upon us in a vivid manner by the noteworthy voyage of our splendid fleet of sixteen battleships around the world, when they were attended by a heterogeneous fleet of foreign merchant vessels acting as colliers and supply ships. In time of war it might be extremely difficult to buy from foreign nations a sufficient number of such vessels to adequately support our navy, and this resort, even if possible, would probably prove extremely expensive.

No less important than the need of American merchant vessels suitable for scouts, colliers, and transports in time of war, is the need of a large and efficient naval reserve with which to man the fleet if occasion required. This feature of naval strength is one which is given great attention by other nations which have first-class navies, and it is a constant source of wonder and surprise to such nations that this prime necessity is neglected by the United States.

PROPOSED LEGISLATION.

Recent attempts to enact suitable legislation to upbuild our merchant marine, to promote our rapidly-increasing commerce with foreign nations, and to provide an adequate naval reserve for our navy, have been meeting with increasing measures of success. Only last year an act providing for increased mail subsidies to ships of 16 and 14 knots running to South America, the Philippines, China, Japan and Australia, passed the Senate by a unanimous vote, and was defeated in the House by the small margin of only three votes. Congress convenes again this month, and two bills have already been introduced into the House; one known as the Greene bill and the other as the Humphrey bill.

The Greene bill is a direct subsidy measure providing for the payment on each entry of a vessel of the United States not exceeding sixteen entries in any consecutive twelve months at the rate of $1 per 100 gross tons for each 100 nautical miles sailing outward and homeward bound, and also providing that owners, after receiving this compensation, are to go bond that they will within two years next after giving of such contract in good faith for the building in the United States of a new

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vessel or vessels of an aggregate gross tonnage at least equal to 25 percent of the tonnage then existing on which compensation is claimed.

The Humphrey bill, on the other hand, is a mail subsidy measure, authorizing the Postmaster-General to pay for ocean mail service, under the act of March 3, 1891, in vessels of the second class, on routes of 4,000 miles in length outward voyage to South America, the Philippines, Japan, China and Australia, the rate per mile not exceeding the rate applicable to vessels of the first class, as provided in said act, and in vessels of the third class on said routes at a rate per mile not exceeding the rate applicable to vessels of the second class in said act, provided that the total expenditure for foreign mail service in any one year shall not exceed the estimated revenue therefrom for that year; sea-going steel steamers of 5,000 gross tons or over to engage only in trade with foreign countries or the Philippines, and wholly owned by citizens of the United States, may be built anywhere and registered, according to this act; but unless such ships are built in the United States they shall not be entitled to mail compensation as provided in this act.

It will be seen that the two foregoing measures are essentially subsidy measures, and since subsidy, either direct or in the form of mail compensation, has been the principal means proposed for the rehabilitation or our merchant marine, it becomes necessary to investigate to what extent and with what success subsidy is employed by other nations, and to determine, as far as possible, how well this means will accomplish the desired end in our own case, and, finally, to weigh carefully the possibilities of other methods which might be available.

AS TO SUBSIDIES.

First, as to the extent to which foreign nations are employing subsidies, the following abstract from the annual report of the Commissioner of Navigation for the fiscal year ending June 30, 1909, shows that eighteen foreign nations are paying annually the sum of $46,896,700 for mail subsidies, Admiralty subventions and navigation bounties as follows:

France	$13,423,737
Great Britain and Colonies	9,089,384
Japan	5,413,700
Italy	3,872,917
Spain	3,150,012
Austria-Hungary	2,984,530
Germany	2,301,029
Russia	1,878,328
Norway	1,102,143
Netherlands	886,011
Sweden	277,752
Denmark	145,000
Belgium	55,970
Portugal	50,000
Chile	253,195
Mexico	75,000
Egypt	54,512
Brazil	1,300,000

Also, that during 1908 the United States paid for the carriage of our ocean mails in American steamers $1,467,255, and to foreign steamers $1,228,032.

The foregoing figures are sufficient to show that the practice of subsidizing lines is practically universal with all maritime countries. In fact, as one writer has aptly expressed it, "Subsidy to shipping in some form or degree, either in the form of payments to regular mail lines, or to all ocean-going ships, is now as fixed a practice as is the use of the gold standard among progressive nations." China and the United States alone hold aloof from this policy, with the natural result that neither China nor the United States has a merchant marine engaged in foreign trade that is worth the name.

Whether or not subsidies are effectual in building up a merchant marine, the reader may judge from the fact that since 1890 British tonnage has doubled, German tonnage has trebled, and Japanese tonnage has increased tenfold. It is true that subsidies, as usually granted in the form of compensation for carrying the mails or as Admiralty subventions, benefit directly only the principal mail and passenger steamship lines, but indirectly a stimulus is given to the shipping of all kinds.

It is not, however, due to any doubt that subsidies will fail to build up the merchant marine in a substantial manner that this method is so stubbornly opposed. The chief objection is the time-worn argument that it is taxing the entire people for the benefit of a few. Why this should be such a serious objection in the case of ship subsidy we fail to see, since the same thing is being done on a far greater scale with the improvement of our rivers and harbors and with our land reclamation projects. As a matter of fact, from the point of view of promoting commerce with foreign nations, ship subsidies would benefit a greater number of people than either of these other undertakings; while from the point of view of furnishing our navy with an adequate reserve force of men and ships, the cost in the form of taxation should be considered no more seriously than is the cost of the navy itself.

OTHER REMEDIES AVAILABLE.

Two other methods for upbuilding the merchant marine have been proposed and defended with considerable vigor, one of which is the free-ship policy. The adherents of this policy maintain that if we were free to build our ships in foreign yards, where the cost of construction is so very much less, we would have no trouble in operating them at a profit.

No greater fallacy than this could be put forward. American capitalists who are now interested in steamship lines operated under foreign flags state that if American registry should be granted their foreign-built ships, they could not afford to take advantage of it, since they are able by maintaining their foreign registry to operate the ships at a lower cost, and also to take advantage of whatever subsidies the nations under whose flag the ships sail are willing to offer. If any further argument is needed to explode the free-ship theory, it is simply necessary to point to the experience which other nations have had with this policy. It has been given a thorough trial by such important maritime nations as Germany, France and Japan, and in each of these cases it proved a dismal failure, the total tonnage increasing very slightly and no stimulous to shipping in general being apparent. Each of these nations has unhesitatingly abandoned this policy for the more effective method of subsidizing.

The third method proposed is one which has already been tried in this country with excellent results. The very first act passed by the first Congress of the United States in 1789 provided thorough protection for American shipping by discriminating duties and tonnage taxes in both the direct and indirect trade. Under the stimulous of this protection our merchant marine grew until it outclassed that of any other nation, over 90 percent of our foreign commerce being carried in American ships. The chief objection to this method to-day, however, is the fact that we have some forty trade treaties with foreign nations, which prohibit any action of this kind. These treaties, our statesmen tell us, are so binding as to preclude the possibility of using this method.

The Shipping League of Baltimore.

Formal organization of the "Shipping League of Baltimore," which was instituted at a dinner given by Mr. Bernard N. Baker at the Maryland Club, October 13, was effected the following day at a meeting of the executive committee in the

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Equitable Building, where the league will maintain its own regular offices.

Composing the committee are: Mr. Baker, former president of the Atlantic Transport Company and Baltimore Trust Company, and president of the league; vice-president, Robt. Ramsay, grain exporter; T. H. Bowles, president of Baltimore Trust Company; Norman James, lumber exporter; Waldo Newcomer, president National Exchange Bank; B. H. Griswold, Jr., of Alex Brown & Sons, and Lynn R. Meekins,

At first adopting a "declaration of principles," the committee authorized Messrs. Baker and Ramsey to draw up a plan of organization and defining the requirements of membership. Besides the sum of $15,000 subscribed at Mr. Baker's dinner, $5,000 additional has been subscribed, thus giving the league all the funds it needs for the present.

Briefly stated, the league will take a foremost position in whatever is done to build up American shipping, with the support of the administration at Washington. The main thing is to arouse every member of Congress in the South to the importance of developing the ports of the South.

PRESIDENT TAFT'S VIEWS ON THE MERCHANT MARINE.

President Taft's views on the upbuilding of the American merchant marine were clearly and forcefully expressed in a speech at Seattle on Oct. 1, an abstract of which is given below:

"We maintain a protective tariff to encourage our manufacturing, farming and mining industries at home and within our jurisdiction, but when we enter into competition upon the high seas in trade between international ports our jurisdiction to control that trade, so far as the vessels of other nations are concerned, of course, ceases, and the question which we have to meet is how, with the greater wages that we pay, with the more stringent laws that we enact for the protection of our sailors, and with the protective system making a difference in the price between the necessaries to be used in the maintenance of a merchant marine, we shall enable that merchant marine to compete with the marine of the rest of the world.

"This is not the only question, for it will be found upon an examination of the methods pursued in other countries in respect to their merchant marine, that there is now extended by way of subsidies by the various governments to their respective ships upward of $35,000,000, and this offers another means by which in the competition the United States ship is driven out of business, and finds itself utterly unable to bid against its foreign competitors. Not only this, but so inadequate is the American merchant marine to-day that in seeking auxiliary ships with which to make our navy of offense or defense, or indeed in sending around the world a fleet, we have to call on vessels sailing under a foreign flag to carry the coal and to supply the other needs of such a journey.

"Were we compelled to go into a war to-day our merchant marine lacks altogether a sufficient tonnage of auxiliary unarmed ships absolutely necessary to the proper operation of the navy, and were war to come on we should have to purchase such vessels from foreign countries, and this might, under the laws governing neutrals, be most difficult.

"The trade between the Eastern ports of the United States and South America is a most valuable trade, and now equals something like $250,000,000; but European nations, appreciating the growing character of this trade, have by subsidies and other means of encouragement so increased the sailings of large and well-equipped vessels from Europe to the ports of South America as visibly to affect the proportion of trade which is coming to the United States by the very limited service of a direct character between New York and South American ports.

"I need not tell you of the inadequacy of the American shipping marine on the Pacific Coast and the growing power for commercial purposes in this regard of the Empire of Japan. Japan is one of the most active and generous countries in the matter of subsidies to its merchant marine that we have, and the effect is only too visible in an examination of the statistics.

"For this reason it seems to me that there is no subject to which Congress can better devote its attention in the coming session than the passage of a bill which shall encourage our merchant marine in such a way as to establish American lines directly between New York and Eastern ports and South American ports, and between our Pacific Coast ports and the Orient and the Philippines.

"We earn a profit from our foreign mails of from $6,000,000 to $8,000,000 a year. The application of that amount would be quite sufficient to put on a satisfactory basis two or three Oriental lines and several lines from the East to South America. Of course we are familiar with the argument that this would be contributing to private companies out of the Treasury of the United States, but we are thus contributing in various ways on similar principles in effect both by our protective tariff law, by our river and harbor bills and by our reclamation service. We are not putting money in the pockets of shipowners, but we are giving them money with which they can compete for a reasonable profit only with the merchant marine of the world.

"From my observation I think the country is ready now to try such a law and to witness its effect in a comparatively small way upon the foreign trade of the United States. If it is successful, experience will show how the policy can best be expanded and enlarged, and the American commercial flag be made to wave upon the seas as it did before our Civil War. It is true that our foreign trade is great and increasing, and this without the merchant marine, but it is also true that the ownership of a merchant marine greatly enhances the opportunities for extending trade for the merchants of the country having such a merchant marine."

"AMERICAN SHIPS AND THE WAY TO GET THEM."

One of the ablest expositions of the views of the adherents of the ship subsidy policy for the upbuilding of the American merchant marine appeared under the above title in the October number of the *Atlantic Monthly*, from the pen of Winthrop L. Marvin, formerly secretary of the Merchant Marine Commission.

Regarding previous protective measures for American shipping, Mr. Marvin says:

"The first Federal Government in 1789 had found the American merchant marine almost as shrunken and dead as it is now—a mere skeleton of 123,000 tons, capable of carrying only a fraction of our commerce, which was conveyed as now largely by British shipping. But the statesmen of 1789, in their very first tariff act, 'for the protection and encouragement of manufactures,' embodied stalwart protection for American ships and sailors through the form of discriminating tonnage and customs taxes, which compelled American merchants to employ the ocean carriers of their own country—and the law required that these ocean carriers should be built in the United States.

"This bold protective measure, which Washington and Madison joined in framing and enforcing, proved so successful that by 1800 our registered merchant fleet had expanded to a

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tonnage of 667,000, carrying 89 percent of our imports and exports, and by 1810 to a tonnage of 981,000, carrying 91 percent of our imports and exports. These policies of ship protection, though modified here and there in the years that followed, were not entirely withdrawn against Great Britain, our chief competitor, until 1849, and by that time they were reinforced by a generous system of mail subsidies, which rapidly developed steamship-building and engine-building in the United States, and gave to our ocean steam fleet a growth in quantity and quality far superior to that of the United Kingdom. These early American mail subsidies, by the way—it is worth recalling now—had been granted by Democratic Congresses, on the recommendation of Southern Democratic Presidents. They created several American steam lines to Europe, with which the feebler and slower British subsidized ships could not compete, and other lines to the West Indies and in the Pacific Ocean.

"The American merchant marine, as it stood at the height of its strength in 1855, when 583,000 tons of shipping were launched in the United States, was the result of a system of national protection deliberately initiated in 1789 by the founders of the Federal Government. Even through these periods, when low-tariff or anti-protection theories had prevailed in Congress and the country, the merchant marine was sedulously fostered by discriminating duties, and later by subsidies to mail lines, while all the time direct bounties were paid to the vessels and men of the deep-sea fisheries—'the nursery of the navy.' There was small protection then for pig iron and cotton cloth, but much protection for ships and, therefore, for shipbuilding. This maritime interest up to 1855 was unquestionably the most progressive, efficient and prosperous interest in America.

"The American merchant marine had prospered and grown amazingly under national protection, and it began to shrink as soon as that protection was withdrawn.

"Every Republican President since Grant has earnestly recommended the upbuilding of the merchant marine, through the form either of mail subsidies to regular lines or of subsidies to the whole body of our ocean shipping. McKinley and Roosevelt were especially insistent on a subsidy policy, and under the administration of President Harrison something was actually done—the enactment of an ocean-mail law which has stood to the present time, and has created the one American steamship line to Europe and excellent lines to the West Indies, Mexico and near ports of South America. But this legislation of 1891 was not liberal enough to sustain steamship lines to the farther and principal South American countries and across the Pacific Ocean."

Mr. Marvin outlines the present situation as follows:

"Here, in a nutshell, is the problem of the American merchant marine. We have established a protective system, and we have left out of that system the industry of the ocean shipowner. We have thereby killed that industry, exactly as we would have killed the manufacture of cotton goods or woolen goods if we had left that industry alone out of the protective system. The manufacturer could not buy his labor and materials in a protected market, and yet sell his product under terms of free-trade competition with all the world. The shipowner has not been able to buy his labor and materials in a protected market—it is only of recent years that materials have been free—and yet sell his product, which in this case is the service of his ship, under terms of free-trade competition with all the world; or, worse, under terms of free-trade competition, frequently aggravated by the bounties or subsidies of other governments.

"On all of the important routes of the world's commerce, the dominating factors in transportation at the present time are the great national mail-subsidized lines of foreign governments."

Regarding the relative efficacy of the "free ship" policy and the ship subsidy policy, Mr. Marvin has this to say:

"It is simply paltering with a great and vital national question to plead that a 'free ship' policy—that is, the purchase of American ships in British yards—would of itself enable American ship owners to meet the conditions with which they are confronted in the Pacific and Atlantic Oceans.

"Germany in the beginning tried the 'free ship' expedient alone, having no shipyards in which either merchant craft or men-of-war of large size could be constructed. The experiment was a complete and acknowledged failure, the German mercantile tonnage, increasing only from 1,098,000 in 1873 to 1,243,000 in 1881. Then Bismarck appealed to the Reichstag for a positive and liberal policy of State aid through mail subsidies, preferential railroad rates, and other potent forms of imperial encouragement. Now the real growth of the German merchant marine began, and the tonnage of the Empire rose to 2,650,000 in 1900, and to 4,232,000 in 1908.

"The experience of France was similar. After a long and patient trial of 'free ships,' the French people found themselves in 1881 with actually a feebler ocean fleet (914,000 tons) than they had possessed in 1870 (1,072,000 tons). In sheer desperation at the utter failure of the 'free ship' experiment, the French government resorted to subsidy and bounty on an extensive scale. The records of the Bureau Veritas show that the French mercantile marine, which was 914,000 tons in 1881 has actually doubled to 1,952,000 tons in 1908—the later increase consisting chiefly of steamships of high character.

"But perhaps the most striking recent example of the success of State aid in the creation of an ocean shipping is the experience of Japan. There, too, the first reliance was placed on a 'free-ship' policy, and there, as elsewhere, while depended on alone, this ignominiously failed. In the war with China in 1894, Japan found herself with only about 200,000 tons of ocean vessels, and with almost no facilities for repairing, not to say building, them. The Japanese statesmen thereupon launched out upon the most generous and comprehensive system of subsidy and bounty, encouraging both 'tramp' ships and regular lines, and developing native shipyards by the expedient of granting a bonus for every ton of ocean shipping constructed. In ten years the Japanese merchant marine had grown from the 200,000 tons of 1894 to 830,000 tons. The total for 1908 is 1,243,000 tons, and the Japanese payments for subsidy and bounty, exclusively to Japanese ships, are not far from $6,000,000 a year.

"China and the United States are the only important governments which have held aloof from the modern policy of direct and liberal national aid to the merchant marine. Subsidy to shipping in some form or degree—in the form of payments either to regular mail lines or to all ocean ships—is now as fixed a practice as is the use of the gold standard among progressive nations.

"This does not mean that the policy of 'free ships' is totally discredited and abandoned: it is simply condemned as insufficient in itself without some form of direct protection and encouragement to native shipbuilding and to navigation. As a rule, the governments which grant subsidy or bounty also allow their people to purchase foreign-built ships, but such ships are usually excluded from the benefit of a part or all of the subsidies, and especially is it required that the faster steamships, the auxiliary cruisers, of the national mail lines shall be of native construction. This, as has been said, is the policy of Germany, and in British mail contracts like that of the Cunard line it is stipulated that the subsidized ships shall be 'built in the United Kingdom.' Unless it be China, or perhaps Russia, no nation now adheres to an absolutely unrestricted 'free-ship' policy, with no thought of native shipbuilding."

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navigation, about $9,500,000 (£1,950,000); Germany, for mail service, $3,000,000 (£616,000); Russia in postal regulations, $2,000,000 (£411,000); Japan, in subventions, $6,200,000 (£1,273,000); Italy, in subventions, $2,700,000 (£554,000); while the United States pays only for the carriage of her mails, about $1,600,000 (£329,000).

Due to the withdrawal, in 1907, of the Oceanic Steamship Company, of San Francisco, of its line to Australia, the American flag has vanished from the commercial routes of the South Pacific.

A most potent argument in favor of the upbuilding of the merchant marine, and one which has had great weight in influencing Japan in this respect, is the use of such a fleet as an auxiliary to the navy. At present the navy is forced to employ foreign vessels to keep it supplied with fuel and other necessities when it leaves our own shores.

I do not believe that we should follow the French method; ships can be put on the ocean and navigated from one port to another if their expenses are paid, but this nursing is without any sound economic policy to back it up; I believe it gives no ultimate benefit to the commerce of the nation adopting it.

In my belief the first duty of the United States is to establish, by mail subventions, permanent lines of communication between her ports and the principal ports of the world, especially those where our products would be most likely to find a permanent market. The character of the service should be clearly stated, and bids for the service required should be asked from responsible shipowners and awards made to the lowest responsible bidder.

Compared to what we annually expend on our navy the expense of such a fleet of merchant auxiliaries would be small. These lines would build up a commerce for this country worth many times what it would cost to maintain them. The method of letting out the subsidies, as it were, to the lowest bidder would secure the required service at the lowest feasible figure, and would be fair to all concerned.

No. 10—Material Handling Equipments for Lake Vessels.

BY RICHARD B. SHERIDAN.

Iron ore was first discovered in the Lake Superior country in 1844, but on account of its location and the almost insurmountable difficulties of handling and transportation, it was nearly twenty years before the ore of this district could be commercially worked.

In 1855 the Government completed its first system of locks at Sault Saint Marie, and the first cargo of ore (114 tons) was carried down the lakes from the mining district. The entire shipments for the year amounted to 1,449 tons, and it was not until 1873 that a season's shipment amounted to more than 1,000,000 tons. In 1907 it reached a maximum of over 42,000,0000.

In the development of this traffic the first step was the construction, in 1860, of a loading dock at Marquette. This dock has been the model on which all of the great ore docks of the Northwest have been built. It consisted of a long line of pockets on a dock extending out into the lake, and arranged so that boats could be brought alongside for loading. The material was brought over the pockets on railway cars, arranged so as to be dumped into the same. Each pocket was equipped with spouts and gates, and of such height as to permit of a slope to the chute, so that the material would run out by gravity. Records are given in which boats of over 10,000 tons capacity have been loaded in as short a time as one hour at modern docks of this type.

The question of loading has not, however, been the perplexing factor in the development of the lake traffic. The great and all-important difficulty that had to be met and overcome was the question of dispatch in the unloading of boats in the lower lake ports. In 1879 a machine was constructed in the harbor of Cleveland, but it had so many faults that it could hardly have been called a success, and it was not until 1880 that the first successful unloading machine was developed, and this date marks the beginning of the real and speedy development of material-handling machinery as well as the improvements in the ship construction of the Great Lakes.

This first device was a cableway machine, built and erected at Cleveland, Ohio, in 1880, under Mr. Alex. E. Brown's supervision, and with its introduction came the formation of the Brown Hoisting & Machinery Company.

This type of machine was an infinite improvement over the old methods, but it had one fault, and that was, that it could only cover a limited storage pile. The next machines were improved upon, and instead of using the cable type, a structural span bridge of 180 feet in length was employed, and both its supporting piers were mounted on wheels, so that the machine could be moved along the full length of the dock.

Machines of this type came immediately into universal use, and nearly every unloading dock was equipped within a few years.

The problem of reducing the hand labor in the boats became an important factor, and, for several years before the grab-bucket was adopted, attempts were made to work mechanical buckets in coal and similar soft materials. About nine years ago the first successful grab-bucket equipment was erected in Chicago by the Hoover & Mason Company.

In the development of the grab-buckets the old principle which the grabs or clamshells employed for so many years in the handling of dirt and other material has been followed out, and consequently the digging power of the bucket depends to a very large extent upon the weight of the bucket itself, and to get a load it is always conceded that the bucket must be dropped as hard as possible to get an entrance into the material.

The design and principle of working of the Brown bucket are novel. On acount of the motion of the digging blades on most buckets, weight, as mentioned above, plays a very important part in how good a digger the bucket is, but in the bucket I refer to a new principle is brought into play. The blades of the Brown buckets, when open, stand almost in a vertical position, and when closing they retain this vertical position for nearly one-half of their stroke, and their action during this time on the material is one of scraping. By the time they have moved through half of their total stroke they have gathered or scraped together a pile of material between them, and at this point the motion of the blades changes to one of rotation, so that they close under the pile of material gathered together. The bucket is so proportioned that the pile which is gathered equals approximately the capacity of the bucket. This grab, also, by its construction is admirably suited for handling lumpy material, inasmuch as the side plates forming the blades of the bucket are cut away at such an angle that the same is past the angle of friction, so that any material getting in front of the edges of these side plates will not cause the bucket to slide over them, but must, by virtue of the angle of these edges, be pushed ahead or to one side. I believe that this feature of the Brown bucket is not found in any other make.

With the adoption of the grab-bucket came still another change in the general design of unloading machinery. Practically no men were needed in the boat for successful operation, and the hoisting and operating mechanism on the machine was all installed on the moving trolley, and this type of trolley has (as is generally known) become to-day to be recognized as the most efficient and rapid.

With the introduction of the grab-bucket there have been practically but two types of machines developed: One, that employing a bucket suspended by the operating ropes, and the other, a grab-bucket carried on a rigid arm. I will not at-

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tempt to discuss the merits from my standpoint of the rigid-arm or "stiff-leg" machine over the suspended bucket type, but will add that they have become a popular machine along the lake districts.

The outcome of operating buckets of this character was that several of the big shipbuilding companies developed a line of boats built especially for the ore traffic, with hatches spaced about 12 feet centers, and clear holds.

The coal trade is by no means a small factor in the total amount of material handled on the Great Lakes during the season. The coal-unloading machines have all passed through the same lines of development that the ore plants have, and, as may be readily supposed, the grab-bucket is as universally used in the handling of coal cargoes as it is in the iron ore traffic. The type of machines employed is very similar to that used for the handling of iron ore, and is designed to handle grabs having a capacity generally of about 2 tons of coal. The question of loading coal into vessels has received during the past ten years considerable attention. A large percentage of the coal which is loaded into boats is handled either by car-dump machines or from elevated docks of the same type as those used in the handling of iron ore.

No. 11—Structural Rules for Ships.

BY JAMES DONALD.

This paper is too voluminous to admit of even an abstract within the limits of this report. It contains a complete set of new rules for the construction and classification of ships which the United States Standard Association originally attempted to compile. Before the completion of the work, however, the United States Standard Association was incorporated with the American Bureau of Shipping, so that the new rules were unnecessary. In order, however, that these new rules, upon which a great deal of work had been expended, should not be lost, and that no results should accrue, it was deemed advisable by the writer to present the rules before the society, in order that all those interested in shipping and shipbuilding might have an opportunity to express their opinion upon them. Nothing new is claimed for these rules. The idea has been to arrange what has been found by experience to be the best practice, and what has already been done, in such a manner that it can be easily understood and the design arrived at quickly and easily.

No. 12—Rivets in Tension.

BY ROBERT CURR.

ABSTRACT.

In investigating the value of rivets under tension the author has been unable to find any authority on same. Structural engineers give no value for rivets under tension, and substitute bolts in all cases where tension is met with. On the Great Lakes, rivets in tension are simply considered on the point of shearing the same. The value of a rivet is taken to be equal to the punched materials.

A plan of a 3-foot frame space showing the main frame and web frame as well as the longitudinals and intercostals of a typical Lake cargo vessel is shown, on which are indicated the numerous points where the construction depends for strength upon rivets in tension. The whole object in the case of the web frame is to make a belt of same, and have it throughout in strength equal to its weakest part through a line of rivet holes spaced eight diameters across same; but since there are numerous connections which depend for their strength upon rivets in tension, the value of which are unknown, or at best doubtful, it is evident that the full value of the materials, as placed, is not obtained.

The author concludes that if rivets in tension have no value some other mode of construction is necessary in order to get the value of materials placed in vessels.

No. 13—Strength of Water-Tight Bulkheads.

BY PROF. WILLIAM HOVGAARD.

ABSTRACT

It cannot be said that we have yet reached a satisfactory solution of the problem of the strength of bulkheads. The most striking evidence of this is perhaps the loss of the White Star liner *Republic*, which, according to various accounts, was due ultimately to leakage and breakdown of the bulkheads.

The dimensions of this class of vessels have increased enormously of recent years, and therewith bulkheads have become larger and deeper. It seems, therefore, not unlikely that the rules of the classification societies, which were framed some years ago, may now need a revision in case of larger ships.

In the present paper it is proposed to deal only with the theoretical treatment of the subject. As the theory of the strength of *bulkhead plating* has already, as far as it is developed, been fully dealt with, it is the particular object of the present paper to deal with the strength of *bulkhead stiffeners.*

It is proposed to deal with the simple case of a rectangular bulkhead, stiffened by a set of parallel, equidistant stiffeners of equal and uniform strength. This is indeed the only case which can be solved theoretically, and that only under certain assumptions, which conform more or less imperfectly to actually existing conditions.

The first chapter of the paper deals with the determination of deflection and stresses for simple bending and shearing, while the tension, which may exist in the stiffener as a whole, is neglected.

In the problems dealt with in Chapter II. is included the tension, which exists in a stiffener fixed at the ends to immovable supports in such a way that the ends are prevented from sliding relative to the supports.

The first problem considered in the first chapter is that of a stiffener supported only at the ends, and four cases are considered: (1) The stiffener is free to turn at both ends. (2) The stiffener, by means of brackets, is fixed with both ends vertical. (3) The stiffener is bracketed only at the foot. (4) The stiffener is bracketed only at the top.

Where both ends are free to slide, the maximum bending moments, and hence the maximum stresses, will be smallest when both ends of the stiffeners are bracketed. When the stiffener is bracketed at the foot only the maximum bending moments will be greatest, greater even than if no brackets had been used, and considerably greater than if bracketed at the top only. That bracketing at the foot only is disadvantageous is probably not generally realized.

The second case considered is with the stiffener supported at both ends and at one intermediate point. Three cases are dealt with: (1) When the stiffener is freely supported at all three points. (2) When it is fixed vertically at the ends. (3) When it is fixed vertically at all three points.

There is considerable advantage in preserving the continuity of a stiffener over a point of support placed at the middle, except where brackets are used both at the middle and at the ends, in which case it is immaterial whether the stiffener is cut or not.

Bracketing of a continuous stiffener is of advantage by uniform load, but it is only necessary to bracket the ends. Thus, for instance, a deck beam supported by a stanchion at the middle should always be well bracketed at the ends.

With uniform plus increasing load there is little or no advantage in bracketing. Thus it appears that, as far as bending is concerned, a vertical bulkhead stiffener, supported at a point near the middle may as well be left entirely unbracketed; but here, again, it must not be overlooked that the brackets may be useful in enabling the stiffener to work by tension and in relieving the boundary connections of the bulkhead.

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If the stiffener is broken at the middle point of support, it is practically indifferent whether it is bracketed at the ends or not, unless it is also bracketed both above and below the middle point, in which case there is a considerable gain.

In the second part of the paper two methods are given for determining the strength of bulkhead stiffeners, where tension is considered, in the case of stiffeners efficiently connected at head and heel to fairly immovable supports, and where the deflection is great.

The first method, which is applicable only to uniform load and to a stiffener attached at both ends to immovable supports, free to turn at these supports as if on hinges, consists of finding an expression for the maximum deflection in terms of the pressure which would produce this deflection in an elastic stiffener freely supported at the ends and with no tension along the neutral axis. The maximum deflection is then expressed in terms of the pressure which would produce this same deflection in a perfectly flexible stiffener hinged at the ends. Equating the two expressions for the deflection, a relation between the two pressures is obtained, and assuming that the sum of the two pressures is equal to the total pressure of the water, which is a known quantity, the unknown pressures can be determined.

The second method deals with a formula which takes account not only of the forces acting normally to the stiffener but also of the tension acting along the neutral axis. This formula is a differential equation of the second order, and the solution for it is based on an expansion of the exponential functions in a converging series. When the stiffener is free to turn at the ends, this method does not lead to a practicable result. When the stiffener is fixed vertically at both ends, however, there is no difficulty, and this case is dealt with both for a uniform load and for a combined uniform and increasing load.

No. 14—The Development of the Gasoline (Petrol) Power Boat.

BY E. T. KEYSER.

ABSTRACT.

Early gasoline (petrol) power boats were similar in design to their forerunner, the naphtha launch. The engine was usually placed as far back in the hull as possible, necessitating considerable rake to the shaft, which, together with the squat of the fan-tail stern, kept the boat down to a very moderate speed. Gradually the position of the engines was moved to the center of the boat, decreasing the rake of the shaft and giving a better disposition of weights, and the compromise and torpedo sterns were developed.

The clipper-bowed, fan-tail stern, high-cabined, glass, windowed launch gave way to the trunk-cabin power cruiser, and this in turn to a large extent to the raised-deck boat, and, in larger sizes, to the bridge-decked cruiser. Finally, the steam yacht field was invaded, until to-day the internal combustion engine is the accepted method of power for craft of less than 100 feet over all.

To show the advantage of the gasoline (petrol) engine in the economy of space, small engine-room force, crew's quarters, etc., the author describes in detail an 83-foot cruiser of the compromise stern type, with engines and gasoline (petrol) tanks installed amidships. This boat is not only a concrete example of the comfort, but of the actual luxury, which may be found to-day in a properly-designed motor boat capable of going almost anywhere to the extent of her fuel capacity, and yet which may be run much more economically than a steam yacht affording anything like the same accommodations.

No. 15—A System of Mathematical Lines for Ships.

BY JAMES N. WARRINGTON.

ABSTRACT

A workable system of mathematical lines for ship-shaped bodies should be capable of producing fairness of form, and at the same time should permit a very considerable degree of freedom in design. Fairness may be attained automatically by the forms of the equations, while freedom in design will depend upon the number of optional constants. In the use of equations the mental process of designing consists largely of thinking in constants, and in order that this may be done with a satisfactory degree of perspicacity the constants should have a recognizable significance, each standing for some definite quality of its curve. As the constants are increased in number, thus giving increased freedom in design, they become involved in each other and lose distinctiveness, and the geometrical conception becomes obscure. Facility of interpretation, therefore, is favored by constants few in number and of easily perceived geometrical effect.

In harmony with this view the author deduces equations which permit no more than what is deemed a necessary degree of freedom in design, and the optional constants possess a sufficiently clear definition to permit of ready interpretation.

Equations are given for computing the area ratio R_a, from which the area of any section can be obtained by multiplying R_a by the area of the greatest section, also for computing the breadth ratio R_b of the load water-line, whence the half-breadth of the load waterline at any station can be obtained in absolute terms by multiplying the greatest half-breadth by R_b. Equations are also given for computing sectional coefficients and ordinates of the sections.

The labor of computation is considerable, but each design worked out is applicable to ships of the same type without regard to size or proportion, for the reason that the ordinates are in ratio form.

THE MARINE POLAR MOTOR.

BY HUGO ANDERSON.

The marine polar motor is a direct reversible Diesel motor and involves the same characteristic process of combustion as the ordinary type of Diesel motor. The main idea of the system is to couple to a propelling motor a maneuvering motor into which air under pressure is admitted when maneuvering, but which simultaneously and continually serves as a blowing-out pump for the propelling motor which works on the two-cycle system.

The arrangement of the motor is diagrammatically shown in Fig. 1, where 1 represents four (or two) combustion cylinders working on the two-cycle system, and 2 represents two double-acting maneuvering cylinders with cranks at right angles. When propelling, atmospheric air is sucked into the maneuvering motor through pipe 5. Here the air is compressed to a low pressure and exhausted through pipe 6 into the working cylinders. In these cylinders the air is compressed to 529-558 pounds. The products of combustion are exhausted through the silencer into the air.

When starting, the maneuvering motor is cut off from the atmospheric air and put in communication with the air receiver 9. The air consumed when maneuvering is automatically made up by the air pump 8, which is put in action when the pressure in the receiver has fallen below a certain value.

The fuel oil from the tank 14 is led through the filter 15 to the pumps 16, which force the oil into the fuel valves 13. From the valves the oil is forced into the combustion chamber by air under pressure supplied by the air pump 11. A receiver and pressure adjuster is shown at 12. The thrust-block 4 is made separately from the motor, and astern of the fly-wheel is a flange coupling.

A complete installation includes also a water pump for cooling the cylinders, etc., for which salt water can be used.

The operation of the maneuvering motor is shown in Fig.

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2. The circle, Fig. 2, represents the motion of a crank pin of a maneuvering cylinder. The inlet valve opens at point 1, immediately after the upper dead center, and the admission of air is cut off at point 2, which corresponds to the line *A-B* in the indicator diagram. From 2 to 3 there is a slight falling off in pressure (*B-C*), which is leveled at 3, the long piston uncovering an aperture in the cylinder jacket communicating with the air. As a matter of fact all air is admitted to the motor through a single pipe 7, but for the sake of clearness two pipes 5 and 7 are shown in the diagram. The piston having passed the lower dead center, the pin being then at 4, the aperture in the cylinder jacket is closed. The air is compressed (*D-F*) and afterwards forced into the receiver 6. When maneuvering, the atmospheric air is cut off and air under pressure is admitted from 1 to 2. The indicator card is then H, I, J, D, E, D, F, G.

The characteristic feature of this working arrangement is, that the starting and the combustion take place in the motor *simultaneously*, whereby the transition period is avoided and a quicker maneuver is obtained, as the double moment, or more, is at hand when starting.

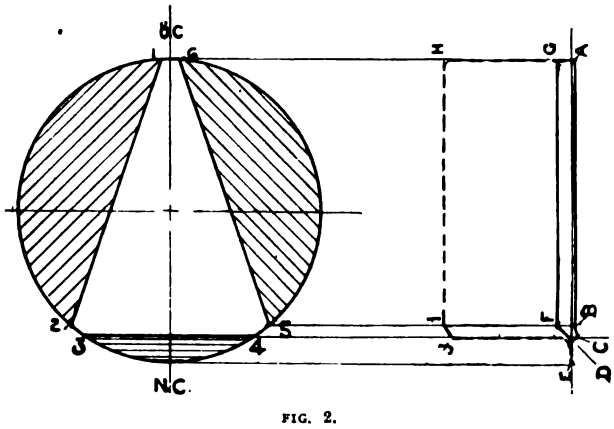

FIG. 2.

A few indicator cards are shown in Fig. 3. *A* is a card from a working cylinder, and it has the same appearance as a card from the ordinary Diesel motor, with the exception of the exhaust line, characteristic of the two-cycle system. *B* is a starting card from the maneuvering cylinder. There are three admission lines, and as many lines representing the air supply to the receiver 6. The highest pressure is 74 pounds. *C* is a card from the maneuvering cylinder when the motor is propelling and taken with a more flexible spring, and finally *D* is a card from the receiver.

As mentioned, one of the regulating parts of the maneuvering cylinders consists of an aperture in the cylinder jacket, uncovered by the piston. This arrangement necessitates the pistons being made very long, a fact which is utilized by disposing air pumps in the interior of the long pistons, special levers for these pumps being in this way saved. Moreover, the pistons are lengthened downwards to form cross-heads.

The operation of the propelling cylinders is seen in Fig. 4; 26 is the receiver, kept under a pressure of about 221 pounds by the maneuvering cylinders; 41 is the washing-out opening, and 42 the exhaust opening; 25 is the fuel valve, the only part of the combustion cylinder that is governed. The valve is regulated in the ordinary way by a cam disc.

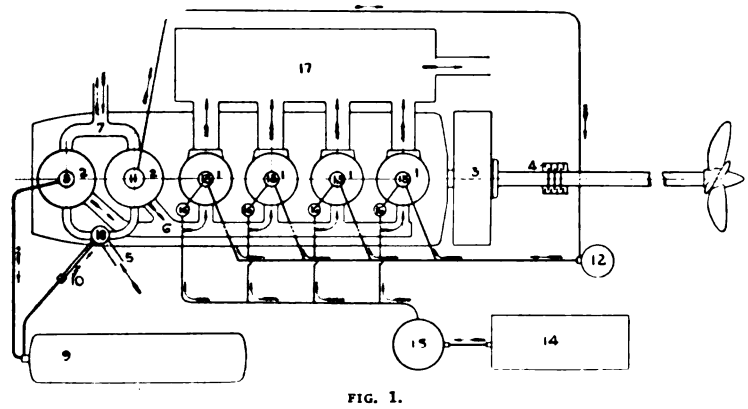

FIG. 1.

The fuel valve is of unique construction, and in particular the carburetor is carefully studied. Thus the same quality of fuel enters the cylinder for large as for small charges, and all the fuel injected by the fuel pump during one stroke enters during the admission period immediately following.

As a result of the special construction of the fuel pump, and of the co-operation between the pump and the fuel valve, and finally of the design of the fuel valve, and specially that of the carburetor, reversal of the motor is accomplished without putting the reversing lever on stop, and also the reversal is accomplished at any speed without risk of damaging or ruining the motor from back explosions and the like.

The reversing of the motor is achieved by altering the position of the fuel pumps and the fuel valves, and this by displacing the cams. To change from ahead to astern, the reversing lever is put in the astern position, at which the fuel pumps are uncoupled first, and then the fuel valves, after the latest delivered fuel charge has been injected and burned in the cylinders. Thereupon the admission of air under pressure in the maneuvering motor is opened. The maneuvering motor is now acting as a pump until the force of the fly-wheel is consumed. When the engine has stopped the maneuvering motor starts to act as a motor and to turn the engine astern. As soon as the engine has started the fuel pumps work again and immediately the ordinary working cycle recommences in the combustion cylinders. There are, then, two turning moments accelerating the fly-wheel, viz., the moment of air under pressure and the moment of combustion. This explains the rapidity of maneuvering.

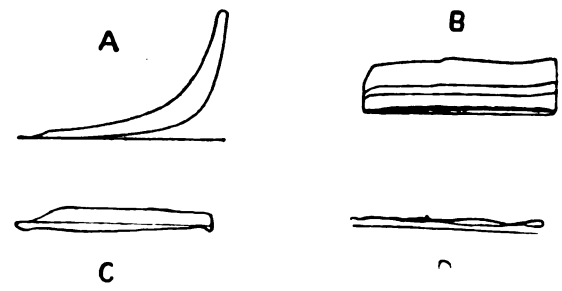

FIG. 3.

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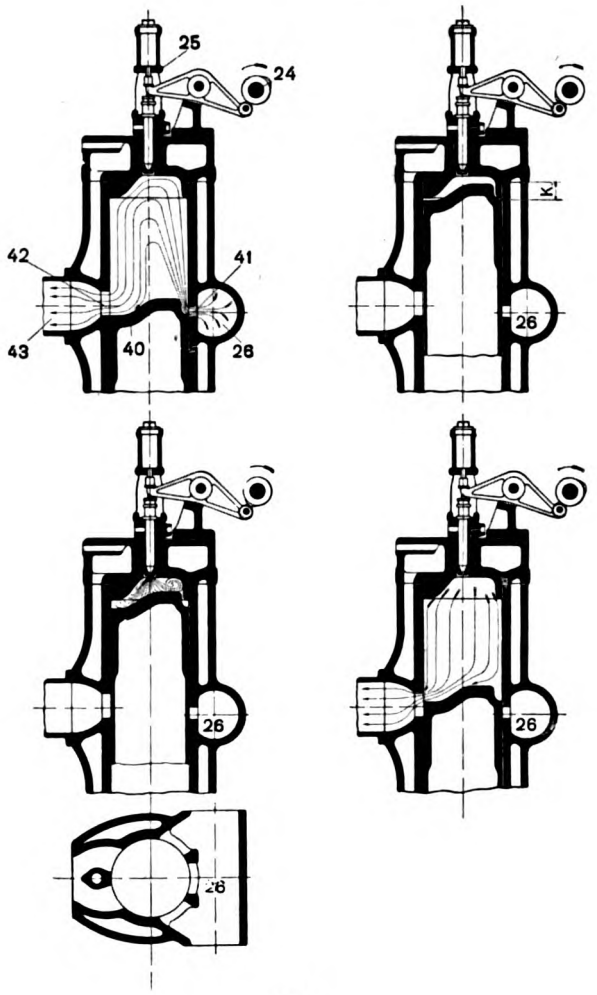

FIG. 4.

Over a year and a half ago the first marine polar motor was installed in a new steel schooner of 300 tons dead weight cargo capacity by the Diesel Motor Company, Ltd., Stockholm, Sweden, the builders of this type of motor. This first motor was of 60-brake horsepower and was originally fitted as an auxiliary, but it has been found economical to keep the engine running constantly when under way in all kinds of weather.

Two 110-brake horsepower motors were also installed in two cargo vessels, the *Rapp* and *Snap*, designed for trading between ports on Lake Vännern, Gottenburg, and the North German Baltic ports. These boats are 105 feet long over all, with a molded breadth of 22 feet 5 inches, a depth of 10 feet 1 inch and a cargo capacity of 300 tons dead weight.

In Fig. 5 is shown the tug *Jakut*, belonging to the oil firm of Nobel Brothers, St. Petersburg, in which are installed two marine polar motors of 160-brake horsepower each. The dotted lines indicate where the old steam machinery was installed, and the saving in space with the oil engines is apparent. It is of interest also to note that the polar ship *Fram*, well known from the polar expedition of F. Nansen, has recently had its steam engine replaced by a marine polar motor of 180-brake horsepower.

Proposed installations for cargo vessels have been worked out to show the advantages of the oil engine. As an example, may be cited a 210-foot cargo vessel with a capacity of 1,400 tons dead weight. The beam of the vessel is 32 feet, the molded depth 15 feet 7 inches, and the load draft 14 feet 3 inches. Propulsion is by means of a marine polar motor of 650 indicated horsepower, giving the ship a speed of 9½ knots. The weight of the propelling machinery is, approximately, 44 percent of the weight of a corresponding steam installation. Forty tons are allotted as the weight of fuel to be carried, and this is sufficient for an uninterrupted run of fourteen days with the oil engines. The same weight of coal would last for a run of three days only. The difference in the weight of fuel means an increase in cargo capacity of not less than 13 percent in the oil-driven ship. With the steam engine it would also be necessary to increase the length of the engine room about 44 percent.

The question of driving the auxiliaries is, of course, an important one, and must be met in an oil-driven ship. In the cargo vessel just described, provision is made for electric lighting by a generator driven by a small Diesel motor. The winches are operated by compressed air from the main engines. The steering gear can also be driven by compressed air. Electricity can be used for heating if its expense is not prohibitive. Heating can, however, be very satisfactorily accomplished by hot water from a small boiler, either fired with oil fuel or incorporated in the exhaust silencer of the motor.

Coming to the cost of operation, in addition to the economy in fuel the use of the oil motor permits a reduction in the engine-room staff, which is a direct economy. In the cargo ship just described, the engine-room staff is reduced by at least one engineer and one oiler, as compared with a steam plant.

In Germany, as is well known, the question of auxiliary sailing ships has been investigated widely, and German auxiliary sailing ships have already successfully competed with tramp steamers. This would seem to indicate that the use of auxiliary sailing ships as cargo carriers can be developed to a much greater extent than is now the case by the use of a motor such as has been described in this article.

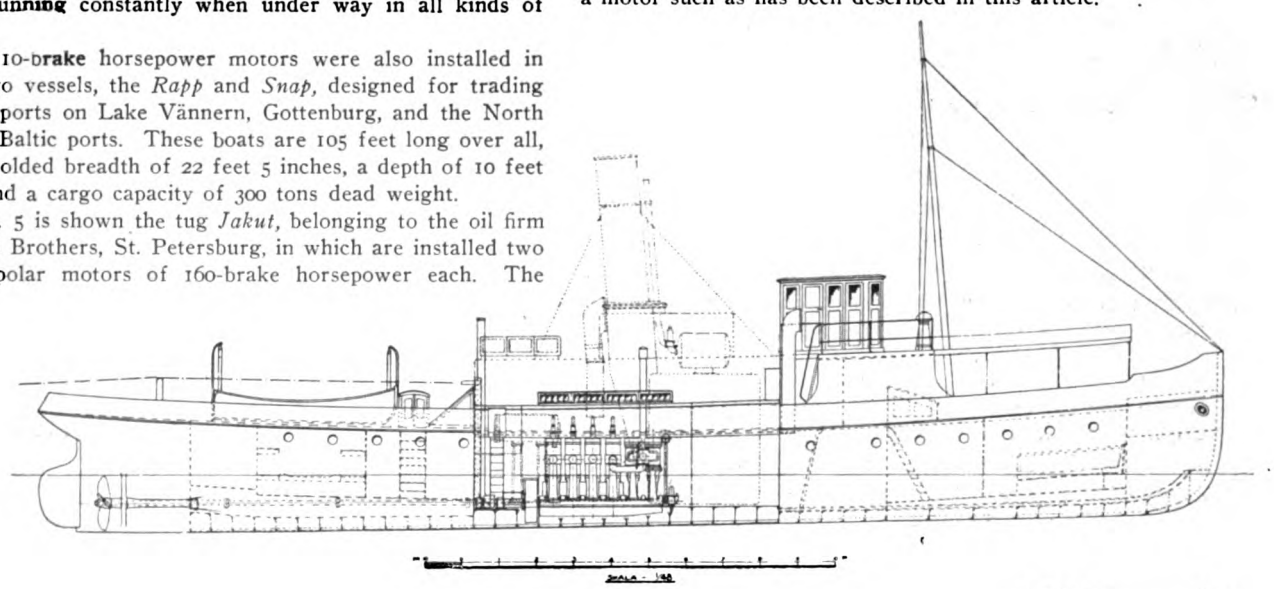

FIG. 5.

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Electric Drive for the Propulsion of Ships.

All proposals for the application of electricity to the propulsion of ships have arisen from an effort to bridge the gap between the economic speed of high-speed prime movers and the relatively low efficient speed of propellers. Since the advent of the steam turbine, which is essentially a high-speed engine, and which gives the best economy in steam consumption at high speeds, the need of some such device has become very urgent, and numerous attempts have been made to design an efficient speed reduction gear which, within the limits of weight, space and cost available in the design of a ship, can be used successfully. Practical solutions of this problem by means of spur gearing and by hydraulic gear are now before the public, and two separate methods of using electricity for this purpose were described by Mr. W. L. R. Emmet at the annual meeting of the Society of Naval Architects and Marine Engineers.

The thing which makes Mr. Emmet's paper of so much value is the fact that it is based not upon mere theories in which numerous loopholes for doubt may be found, but upon actual facts and designs which have been worked out in every significant detail, and which might, if opportunity offered, be contracted for immediately. The author even goes so far as to state that the designs would be fully guaranteed. With such carefully worked-out and thoroughly practical designs as a basis for consideration, these proposals merit the most careful investigation.

Both methods suggested by Mr. Emmet provide for twin screws and, for the sake of comparison, the propeller speed and other details are taken the same as for a direct drive by Curtis turbines, which was the design actually adopted in the ships for which these systems of electric drive were proposed. Since, however, the main object of the electrical apparatus is to introduce a speed-reducing bond between the turbine and the propeller, better results than those set forth in the paper could be obtained by selecting the best propeller speed for the ship and then designing the electric motors to obtain this speed.

In the first arrangement described, which is designated as a combination drive, there is a low-pressure turbine and an electric motor on each propeller shaft, at high speeds the turbine generating about three-fifths of the total power. The motors are operated from generators driven by high-pressure turbines, from which steam is exhausted into the main low-pressure turbines. Under cruising conditions the high-pressure turbines and electric drive alone are used, the low-pressure turbines running idle. Since reversal is obtained by reverse wheels in the low-pressure turbines, the point was raised in the discussion of the paper of whether with the low-pressure turbine running idle most of the time, and consequently cold, it would be possible to reverse instantly without the possibility of serious damage to the turbine. Experience with large stationary turbines of similar design, however, has shown that live steam can be turned into the reverse stages without previous warming up with no ill effects whatever.

This combination plan of turbines and generators was one actually proposed for the new battleships *Arkansas* and *Wyoming*, the main reasons why it was rejected, beyond the natural hesitancy of attempting on so large a scale an absolutely untried and unknown project, we understand to have been the excessive weight of the installation and its high cost, the bids for this arrangement being about $645,000 (£132,500) above the lowest bid for Parsons turbines. As to the weight, Mr. Emmet's figures show that, disregarding piping, bearings, shafting, valves or

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auxiliaries, the combination drive is about 7 percent heavier than that for Parsons turbines alone.

Mr. Emmet brings forward, however, a second method of electrical propulsion, in which each propeller shaft is provided with two motors, one adapted for use at low speeds and the other adapted for high speeds, the generating units in this case being designed to give a very uniform efficiency through wide ranges of load and speed. The reversal in this case is obtained through the motors themselves. This form of drive, as compared with the direct Parsons drive, is shown to be 27 percent lighter in weight, exclusive of piping, bearings, shafting, valves or auxiliaries. Thus it is quite apparent that the disadvantage of excessive weight which exists in the first case can be overcome. The weight of piping for an installation of Parsons turbines with the ordinary arrangement of four shafts is very considerable, since the piping is exceedingly complicated and extensive. Other items, such as the weight of bearings, thrust blocks, valves, etc., would be considerably less in the case of the electric drive than in the case of the direct Parsons drive. Also, it is probable that the boiler weights could be reduced somewhat, since the steam consumption of the turbines would be less when running at their most economical speed.

An interesting possibility was mentioned during the discussion of this paper, and that is the possibility of controlling the ship direct from the bridge if the electric drive were used. There are a number of instances on record where disasters have occurred through a misunderstanding of signals between the bridge and the engine room. Therefore, the advantage of some system of direct control would be inestimable. Since, however, the control of the ship when driven by electricity involves not merely the control of the electrical apparatus, but also the control of the steam turbines which drive the generators, it is hardly to be expected that satisfactory means could be developed for this any more than satisfactory means have been developed for the control of a direct turbine drive. It should be remembered that in the electric drive the variation of speed is obtained by steam and not by electricity, the electrical apparatus forming merely a fixed speed-reducing bond between the turbines and propellers.

Another possibility which undoubtedly would be opened up by the adoption of electric drive is the application of internal-combustion engines as prime movers on board ship. Electric drive, in fact, offers the best, if not the only, solution of this problem at present.

Whether we are justified in assuming that it would be possible to build a large marine power plant of this type without previous experimental installations of smaller power and expect it to succeed, will depend largely upon the faith which individuals have in the ability of the engineering profession to undertake such enormous advances at a single bound. Such steps have frequently been taken, however, particularly in the early days of the steam turbine, as it will be recalled that at the time the turbines were designed for the *Lusitania,* an installation which aggregated 68,000 horsepower, the largest marine steam turbine then built was only 4,000 horsepower. Since we are assured by Mr. Emmet of the practicability of the electrical apparatus, and since his confidence is based upon the fact that motors and generators of exactly the same design which would be used in this case are now being used with good efficiency and unquestioned reliability for such services as mine hoisting, rolling-mill work, etc., where the requirements are far more exacting than they would be for ship propulsion, there does not seem to be much cause for doubting the practicability of the electric drive.

Problems Relating to the Resistance of Ships.

The subject of the resistance of ships furnishes one of the most prolific fields of investigation that is open to naval architects. However settled the design of any type of ship becomes in practice, there is nearly always the possibility of making some change in the form of hull which will diminish the resistance and, consequently, permit either an increase of speed for the same power or a reduction in the power required to attain the designed speed, with a corresponding increase in the dead weight carrying capacity of the vessel. A case in point is the design of the *Monitoria,* which is described on page 477. This vessel is built with two longitudinal corrugations in the shell plating between the load waterline and the bilge, the effect of which is claimed to be an improvement of 5 percent in the speed of the ship over that of a similar ship of the ordinary type at 10 knots.

While such radical improvements as the foregoing are rare, yet the continual investigation of special problems in resistance by means of model tank experiments is steadily increasing the fund of general information which can be brought to bear upon new designs and the improvement of old ones. Particularly valuable in this respect are the researches of Naval Constructor Taylor at the model towing basin in Washington and of Professor Sadler, at the University of of Michigan, and it is gratifying to find among the papers presented at the recent annual meeting of the Society of Naval Architects and Marine Engineers two valuable papers from these authorities bearing upon the general subject of resistance. There may be no way to diminish to any great extent skin or frictional resistance, which in full ships at moderate speeds comprises about 70 percent of the total resistance; but very probably, as in the case of the *Monitoria,* there are forms for ships' hulls yet unknown which will admit of a reduction of that part of the residuary resistance which is due to wave making.

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Progress of Naval Vessels.

The Bureau of Construction and Repair, Navy Department, reports the following percentages of completion of vessels for the United States navy:

	Tons.	Knots.		Oct. 1.	Nov. 1.
BATTLESHIPS.					
S. Carolina..	16,000	18½	Wm. Cramp & Sons.....	99.0	99.9
Delaware ...	20,000	21	Newp't News Shipbuilding Co.	96.8	97.4
North Dakota	20,000	21	Fore River Shipbuilding Co..	95.2	96.6
Florida	20,000	20¾	Navy Yard, New York.....	33.7	38.3
Utah	20,000	20¾	New York Shipbuilding Co..	44.4	50.0
Arkansas ...	26,000	20½	New York Shipbuilding Co...	0.0	1.0
TORPEDO-BOAT DESTROYERS.					
Smith	700	28	Wm. Cramp & Sons.....	98.4	99.4
Lamson	700	28	Wm. Cramp & Sons.....	91.4	92.8
Preston	700	28	New York Shipbuilding Co...	94.8	95.9
Reid	700	28	Bath Iron Works.....	94.3	100.0
Paulding ...	742	29½	Bath Iron Works.....	36.3	46.1
Drayton	742	29½	Bath Iron Works.....	30.5	37.0
Roe	742	29½	Newp't News Shipbuilding Co.	64.6	66.4
Terry	742	29½	Newp't News Shipbuilding Co.	63.9	65.8
Perkins	742	29½	Fore River Shipbuilding Co...	56.0	59.3
Sterrett	742	29½	Fore River Shipbuilding Co...	53.4	58.1
McCall	742	29½	New York Shipbuilding Co...	29.6	34.2
Burrows	742	29½	New York Shipbuilding Co...	29.2	34.2
Warrington..	742	29½	Wm. Cramp & Sons.....	47.3	53.0
Mayrant	742	29½	Wm. Cramp & Sons.....	51.5	54.6
No. 32.....	...	....	Newp't News Shipbuilding Co.	1.1	3.2
No. 33.....	...	....	Bath Iron Works.....	4.0	7.9
No. 34.....	...	....	Fore River Shipbuilding Co...	3.2	5.2
No. 35.....	...	....	Fore River Shipbuilding Co...	5.3	8.2
No. 36.....	...	....	Wm. Cramp & Sons.....	1.7	3.4
SUBMARINE TORPEDO BOATS.					
Stingray	...	..	Fore River Shipbuilding Co...	99.1	100.0
Tarpon	...	..	Fore River Shipbuilding Co...	99.1	100.0
Bonita	...	..	Fore River Shipbuilding Co...	99.0	100.0
Snapper	...	..	Fore River Shipbuilding Co...	99.0	99.3
Narwhal ...	...	..	Fore River Shipbuilding Co...	98.9	100.0
Grayling	...	..	Fore River Shipbuilding Co...	98.5	100.0
Salmon	...	..	Fore River Shipbuilding Co...	86.7	87.0
Seal	...	..	Newp't News Shipbuilding Co.	26.2	28.2
Carp	...	..	Union Iron Works.....	6.0	16.0
Barracuda ..	...	..	Union Iron Works.....	6.1	16.1
Pickerel	...	..	The Moran Co.....	10.7	13.0
Skate	...	....	The Moran Co.....	10.6	13.1
Skipjack	...	..	Fore River Shipbuilding Co..	7.6	8.4
Sturgeon ...	...	..	Fore River Shipbuilding Co..	7.6	8.4
Tuna	...	..	Newp't News Shipbuilding Co.	3.8	7.7

ENGINEERING SPECIALTIES.

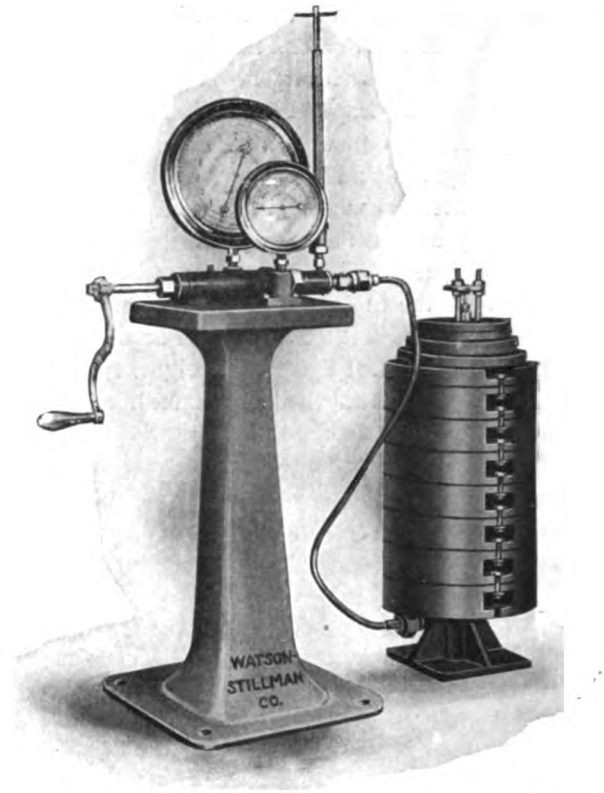
Improved Calibrating Apparatus for Hydraulic and Other High-Pressure Gages.

The bursting of machine parts and fittings from excessive fluid pressure is usually accompanied by considerable danger, expense and delays for repairs. For this reason pressure gages should be calibrated at regular intervals. Under the higher hydraulic pressures it is frequently the case that the same gage will show different percentages of error at different pressure readings, and these can be compensated for in ascertaining the true pressure only by comparing with a "master" gage of known accuracy or by loading with a known pressure.

The outfit which we illustrate performs these two functions of testing by comparison with a master gage and of testing the accuracy of the master.

For the first only the part on top of the stand is required. This consists primarily of a cross made of hydraulic bronze. The gage being tested, which may register any pressures up to 16,000 pounds per square inch, and the master are attached to the front and back ends of the cross respectively. At the left is a crank-operated screw displacement piston, by means of which the desired pressure may be produced within the pressure chamber. A suitable stuffing-box prevents leakage past the piston. To the right end is connected a stop valve and filling cylinder. This permits (1) some of the liquid to be withdrawn from the pressure chamber before removing the gage being tested and (2) filling after a gage is put on. There is thus no danger of spilling the oil.

For testing the master gage, the special weight-loaded, hardened and ground steel piston and cylinder are attached at the right by means of flexible copper tubing, as shown. These parts are cut out by a stop valve when not testing the master. The cylinder is long enough to have the center of gravity of



the weight below the center of support. When the weights are revolved the friction due to lifting the weights is practically eliminated.

The aparatus is made by the Watson-Stillman Company, of New York.

Crude Oil Engine for Marine Purposes.

Road & Rail Engineering, Ltd., of Duffield, near Derby, have just placed on the market an internal combustion engine, known as the Duffield crude oil engine, which is designed particularly for use as auxiliary power for small and medium-sized coasting vessels, where the main requirements are small initial outlay, low cost of fuel, ease and simplicity of operation, elimination of dangerous volatile fuels and accessibility for adjustment.

Small capital outlay is necessary, since such engines are used, perhaps, only 2 or 3 hours a day. This has been secured by arranging for a normal speed of 600 revolutions per minute, both for the 25 and 50-horsepower sizes, keeping the maximum pressure within normal limits, and making all parts as plain and simple as possible. The fuel used is crude, unrefined paraffin (kerosene) oil, such as is used for the enrichment of gas in gas works, and which can be bought at almost any English port for half the price of ordinary refined paraffin (kerosene). This fuel can be used with success in this engine, since by means of a novel form of vaporizer the troublesome "hard base," which ordinarily forms tarry deposits in the cylinder and clogs the valves, is entirely removed before the explosive mixture enters the cylinders. The oil is injected under pressure into a form of atomizer and is thrown in the form of a finely divided spray into a cylindrical chamber, which is heated externally by the exhaust gases. Thus that portion of the oil which is suitable for use in the cylinders is vaporized and drawn together with the necessary air into the cylinders. The "hard base," which will not vaporize at ordinary temperature, is thrown against a collector and runs down a drain. At starting, the same atomizer as is used for running is used to blow the flame

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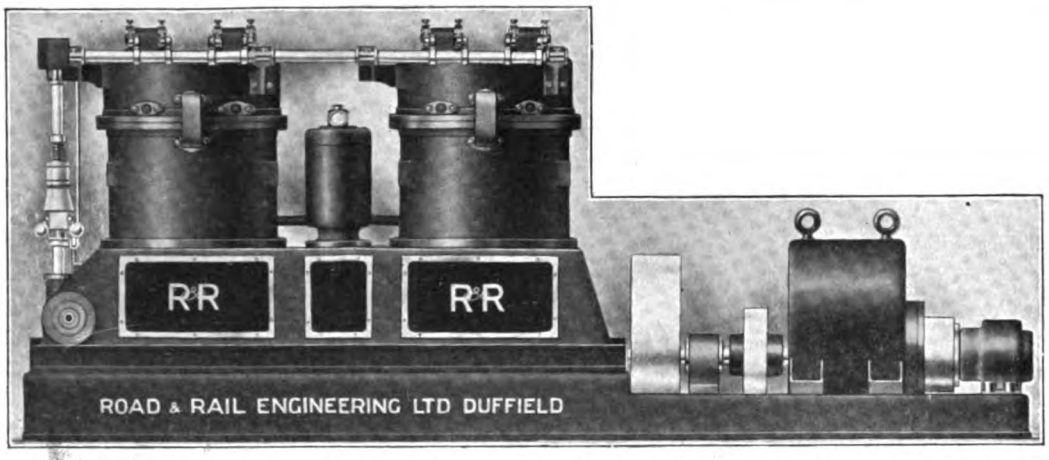


through the vaporizer for the preliminary heating. Air is admitted during this process by openings near the atomizer, and, by opening a valve at the end of the vaporizer, the flame passes straight through into the exhaust. No petrol (gasoline) or other spirit is required at starting, neither are external lamps necessary.

The system of governing in this engine is important. The valves controlling both oil, air and the resulting vaporized mixture are acted upon simultaneously, and the relative motions are so designed that, it is claimed, the mixture in the cylinders is always such as to give the maximum efficiency at the particular load on the engine. Over 50 percent speed regulation can be obtained almost instantly. Ignition is either by high-tension trembler coil or low-tension magneto.

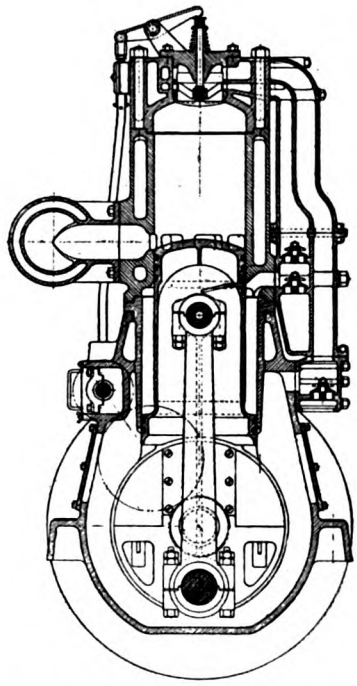
The Straub Marine Two-Cycle Scavenging Engine.

These engines, built by the Page Engineering Company, Baltimore, Md., are of the scavenging type, and resemble in this respect the well-known Korting stationary engines. While the engine is two-cycle in principle, in that an explosion takes place every revolution in each cylinder, the Straub engine otherwise bears scarcely any resemblance whatever to the two-cycle engines as known in the marine trade. In general appearance it is very similar to the four-cycle, having cams and valves, and while it is claimed to be of equal efficiency and reliability, possesses the tremendous advantage of greater power on less weight and space. To this must be added greater simplicity and less first and last cost.

The explosive charge after expansion is completely blown out by a charge of clear air, with perfect scavenging effect, before the fresh charge of explosive mixture is admitted to the cylinder. The admission of the latter is so timed that none of the gas is lost through the exhaust ports. With a clean, cool charge a high compression can be carried with a mean effective pressure equal to the best four-cycle practice.

The Straub scavenging engine was primarily designed to meet the exacting requirements for a successful producer-gas engine, and it is claimed by the manufacturers to be the only engine built in America to-day which successfully fulfills these specifications for marine service. The very elements of design that make for the successful gas engine produce a gasoline engine of equal merit. Both producer gas and gasoline can be used with a change only in compression.

The cylinders and pistons are trunketed, the working, or explosive, cylinder being of the ordinary construction, with the exhaust ports extending around the entire circumference of the cylinder. The exhaust gases pass with the utmost freedom from the cylinder to a large water-jacketed exhaust manifold or receiver, bringing the pressure almost instantaneously to atmospheric, if not, indeed, creating something of a vacuum in the cylinder itself, due to the velocity of the exhaust. The cylinder and piston are enlarged below the exhaust ports, thus forming an annular ring or space, which serves as the gas compressor. Clear air only is compressed in the base, and of a volume about 50 percent in excess of both the working piston displacement and clearance in the combustion chamber, the full diameter of the enlarged piston being effective for this purpose. Both air and gas are admitted to the cylinder through a mechanically-operated inlet valve in the cylinder head. This valve is so controlled that part of the air is admitted first, and completely blows out the burnt gases, not only performing the scavenging function but



also cooling off the igniter, valve, interior of the cylinder and head of the piston. A further opening of the valve admits the gas, which mixes intimately with the air from the base. It is claimed that no back firing is possible, and the charge being fresh and cool a compression of 85 pounds on gasoline and 150 pounds on producer gas can be obtained without pre-ignition.

These engines adapt themselves very favorably to air-starting and reversing, and three-cylinder engines of this type are equivalent in this respect to the six-cylinder of the four-cycle type, same bore and stroke. In fact, the mechanism required for reversing is so simple that the operation can be compared very closely to that of a steam engine with one lever for the throttle and one lever for the reverse. The ignition is

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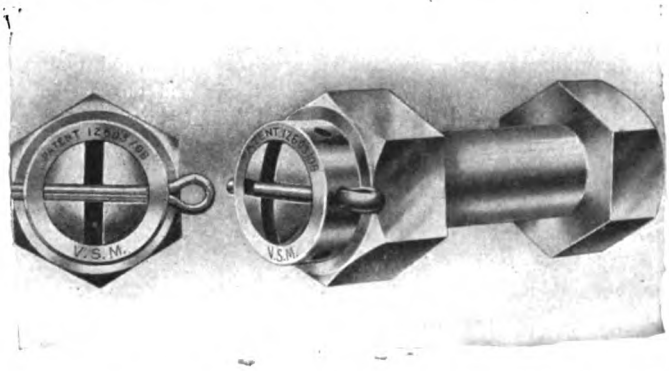
either make-and-break or jump-spark, with magneto-geared or Bosch magnetic make-and-break.

These engines are built in three-cylinder sizes: 7 inches by 9 inches, 2, 3 and 4 cylinders rating on gasoline up to 60 horsepower at 350 revolutions per minute; 9 inches by 12 inches, 2, 3, 4, 6 cylinders rating up to 200 horsepower on gasoline at 300 revolutions per minute; 12 inches by 16 inches, 3 and 4 cylinders rating up to 200 horsepower on producer gas at 225 revolutions per minute.

The V. S. M. "Turret" Lock Nut.

The collar of the "Turret" nut, manufactured by Vickers Sons & Maxim, Ltd., 32 Victoria street, London, S. W., is provided with six or more adjusting split-pin holes, and the bolt with one or two slots, according to its diameter. The holes and the slots combined, as will be seen from the illustration, allow of a fine adjustment; in the 1-inch size, for instance, the lineal movement for each step between nut and bolt would be

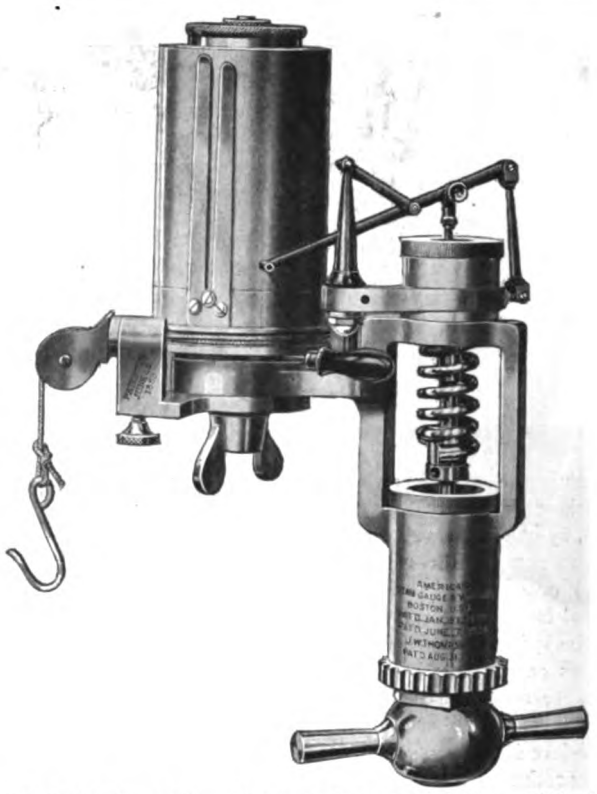


.005 inch. The holes for split-pins and the slot in top end of bolt are always in view, making it easy to locate the alinement of both, and so preventing the workman from overstressing them; further, the slot in bolt can be used with a screwdriver to prevent any turning when nut is being screwed on. This removes any necessity for "feathers" or hexagon headed bolts, and adds to the economy gained by using the device

The American Thompson New Exposed Spring Improved Indicator.

With all the activity which has been stimulated in the indicator field during recent years, it is interesting to note the latest development of the original Thompson indicator, made by the American Steam Gauge & Valve Manufacturing Company, Boston, Mass. This new indicator, shown in the accompanying illustration, is styled by its makers the "American Thompson New Exposed Spring Improved Indicator." Although this new indicator has been in the process of development for some time, it has been but recently placed on the market, yet the success with which it has met, and the commendation it is everywhere receiving, make it worthy of special mention, as it is claimed to be one of the most practical and successful improvements in the development of indicators since the invention of the detent motion.

One of the basic causes for the superior accuracy of the American Thompson indicator, is the short piston rod which the spring surrounds; in other words, the shorter the piston rod, the less the liability of binding, and therefore the greater uniformity and accuracy. This new type of exposed spring is accomplished without increasing the length or weight of the piston rod to any appreciable extent, thus avoiding any increase of inertia in the moving parts; an error of design



liable to be found in outside spring construction. This design is claimed to accomplish all the accuracy originally gained with the short piston rod, with the advantage which all exposed or outside springs possess of being impervious to the effects of heat or cold, and readily changed.

Annual Meeting of the American Society of Mechanical Engineers.

The thirteenth annual meeting of the society will be held in the Engineering Societies' building, 29 West Thirty-ninth street, New York, Dec. 7 to 10.

TECHNICAL PUBLICATIONS.

Machine Shop Drawings. By Fred H. Colvin. Size, 4½ by 6¾ inches. Pages, 139. Figures, 91. New York, 1909: McGraw-Hill Book Company. Price, $1.00 net.

This little book, which is of convenient pocket size, is intended to help those who do not thoroughly understand the reading of drawings rather than as an attempt to teach drawing itself. The first thing an apprentice must learn in any kind of machine work is how to read a drawing. Familiarity with reading of drawings leads easily to the making of the drawings themselves. Many actual examples are given in the book from the drawing-room practice of the leading shops in America and the meaning of each is carefully explained. Some attention is given to laying out work, such as gearing of different kinds, and some hints are also given in regard to sketching.

Slide Valve Motion for Marine Engineers. By Peter Youngson. Size, 7¼ by 9¾ inches. Pages, 132. Figures, 74. Glasgow, 1909: James Munro & Company, Ltd. Price, 5s. net.

Various attempts are frequently made to fill in the gaps between purely theoretical and purely practical training in various lines of mechanical work by means of books treating of theoretical subjects in a practical way. This book comes under this class, and is intended primarily to give an apprentice as good a practical knowledge as possible of the design, construction and handling of various types of valve gear be-

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fore he has an opportunity for actual sea service. Not only will those students who lack practical experience find the book useful, but also seagoing engineers who lack theoretical training will find the book equally interesting and instructive. It is clearly and concisely written and amply illustrated with detailed diagrams. The final chapter includes 100 questions on slide-valve motion which have been selected from Board of Trade examinations. A thorough mastery of this book should give the applicant for a Board of Trade certificate reasonable confidence to approach the examinations relating to slide valves.

Engine Lathe Work. By Fred H. Colvin. Size, 4½ by 6¾ inches. Pages, 180. Figures, 127. New York, 1909: Hill Publishing Company. Price, $1.00 net.

The fundamental principles of the proper running of all machine tools, while well known by the older mechanics, must be learned by apprentices and younger machinists. This book is offered as an aid to such young men, and, therefore, it is not intended to present anything startlingly new in the way of machine-shop practice. The suggestions and methods outlined represent, however, good shop practice; and while they have been written especially for those with a limited experience, it is quite probable that many ideas and suggestions may be new to some of the older men who have not had a chance to see what other shops are doing. Naturally, the first chapter of the book is devoted to a detailed description of the engine lathe. Following this the various operations are described.

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SELECTED MARINE PATENTS.

The publication in this column of a patent specification does not necessarily imply editorial commendation.

American patents compiled by Delbert H. Decker, Esq., registered patent attorney, Loan & Trust Building, Washington, D. C.

920,283. FLOATING DRYDOCK. WILLIAM THOMAS DONNELLY, OF BROOKLYN, N. Y.

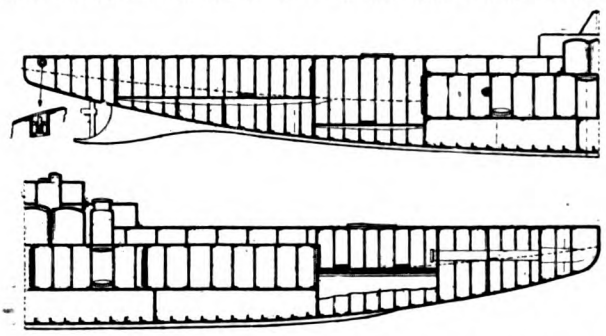
Claim. 2.—In a floating drydock, pontoons comprising transverse beams at top with spaces between them, transverse beams at the bottom with spaces between them, longitudinal beams at the top and bottom, a truss member, diagonal braces, and tension members having plates that are mounted on and bridge the longitudinal beams in the spaces between the transverse beams. Three claims.

925,506. APPARATUS FOR CONDENSING STEAM. DANIEL ARTHUR QUIGGIN, OF LIVERPOOL, ENGLAND.

Claim 1.—In apparatus for condensing steam or for heating or evaporating water by steam, a casing, tube plates, heat transmitting surfaces within said casing and connected to the tube plates, comprising straight or substantially straight tubes having each a trough or troughs extending longitudinally thereof on its upper side, the tubes being arranged with a slope so as to collect and convey to one end of the tubes and casing the drainage resulting from the condensation of the steam on the outer surface of the tubes. Five claims.

926,007. SUBMARINE OR SUBMERGIBLE BOAT. SIMON LAKE, OF BRIDGEPORT, CONN., AND EDWARD LASIUS PEACOCK, OF WEST MOUNT, MONTREAL, QUEBEC, CAN., ASSIGNORS TO THE LAKE TORPEDO BOAT COMPANY, A CORPORATION OF NEW JERSEY.

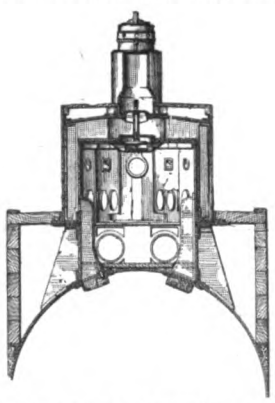
Claim 2.—A submarine or submergible boat, constructed with an intermediate double-compartment section, a substantially semi-circular hull portion arranged below the section, bow and stern sections extending



from the ends of the said double-compartment section and from the said hull portion, the axes of which are upwardly inclined toward the extreme ends of the boat. Fifteen claims.

926,065. SUBMARINE VESSEL. SIMON LAKE, OF BRIDGEPORT, CONN.

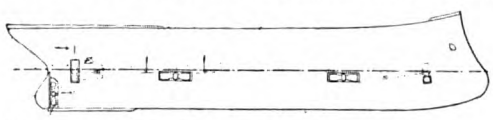
Claim 5.—A submarine vessel, having a navigating turret extending from the hull thereof, air and water-tight compartments arranged in the



turret, one of which forms an air duct which communicates with the hull of the vessel, a valve-controlled opening leading into the duct, and a valve controlling the inner end of the duct. Twenty-eight claims.

928,957. STEERING MECHANISM. LOUIS GABETTI, OF HOBOKEN, N, J.

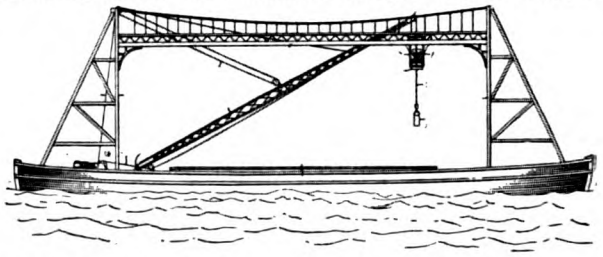
Claim 1.—In a steering mechanism for vessels, a casing formed in the side of the vessel and having pockets therein, said casing opening into the water, a shaft extending through the casing, packing plates



arranged in said pockets, and connected to said shaft, a frame in which the shaft is journaled, a steering propeller arranged in the casing and carried on the shaft and means for shifting said frame. Two claims.

929,139. BARGE. HENRY WILLIAM KIRCHNER, OF ST. LOUIS, MO., ASSIGNOR TO RIVER & RAIL TRANSPORTATION COMPANY, OF GUTHRIE, OKLA., A CORPORATION OF OKLAHOMA.

Claim 9.—The combination with a barge of a plurality of interchangeable boxes, each having fastening devices at its ends, said barge having divisions whose widths are multiples of the space required



by a single box, and the partitions between the divisions of a barge being provided with means for co-operating with the locking devices on the boxes, and with horizontal tracks, and skids on said tracks for supporting said boxes. Ten claims.

British patents compiled by G. F. Redfern & Company, chartered patent agents and engineers, 4 South street, Finsbury, E. C., and 21 Southampton building, W. C., London.

9,679. PROPELLERS. P. ST. G. KIRKE, SCOTLAND.

Claim.—Water enters and is by vanes rotated so that the centrifugal action maintains a constant pressure on its outer walls, whilst its rate of axial flow is not retarded. The rotary direction is changed by other vanes to an axial, rearward direction for propelling the vessel. A smaller similar propeller may be used in running astern and may be combined with the main propeller.

23,981. DREDGING. J. M. CRYER, POULTON-LE-FYLDE, AND C. A. HOLT, BOLTON, LANCASHIRE.

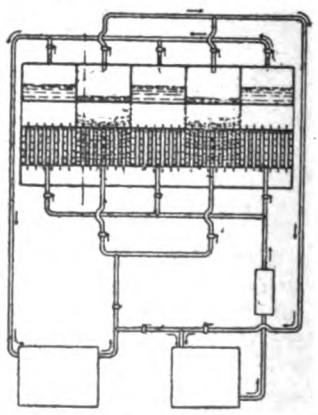
In dredging apparatus for cleaning mill and other reservoirs, one or more portable pontoons are employed, which are adapted to be readily taken to pieces for storage or removal from one reservoir to another. The deck has a baseplate with a footstep for the central post of a crane carrying a dredging bucket. The sludge pump, also located on the deck, may be provided with flexible suction piping having a rose or nozzle at its end.

27,266. SUBMARINES. A. J. KEMP, F. W. RANDALL, SOUTHSEA, AND F. A. PRIMROSE, FOLKESTONE.

Claim.—The fine adjustment of the degree of submersion or immersion of submarines is here effected by simultaneously controlling the admission and discharge of water to or from the auxiliary ballast tanks, a pair of which is provided equidistant fore and aft of the buoyancy center and preferably on opposite sides of the vessel. These tanks have displacers or pistons actuated by screws, which engage a cross-head fast on the piston rod. The screws are turned by wormwheels engaging worms on a shaft, which also similarly operates the other displacer or piston, a portion of the shaft being provided with universal joints or other devices to allow transmission of power from one side of the ship to the other.

26,794. SUBMARINE VESSELS. R. D'EQUEVILLEY-MONTJUSTIN, KIEL, GERMANY.

The steam required to drive the prime mover for the propulsion of submarine vessels is generated in a boiler, fired in the ordinary way, when the vessel is awash; but when the vessel is submerged, the steam is generated in the well-known way in fireless boilers, by means of an absorbent medium, such as caustic soda, caustic potash, or the like. The boiler is divided into compartments, some of the compartments being adapted to receive water and some the absorbent medium. Preferably the compartments are placed alternately. Tubes pass horizontally through the compartments and are fitted with central divisions. Lateral oscillations of the water in the boiler are prevented by a central longitudinal



division. Under the boiler is situated the heating channel for oil firing, from which heating tubes lead through the compartments to the smoke passage. From the compartments steam is led to the engine from which it is exhausted by pipes leading to the compartments and the condensed respectively. The condensed steam is pumped into the boiler in ordinary running; but when submerged, the steam passes through the soda solution in the compartments, and the heat generated is utilized in evaporating the water in the other compartments. When the vessel is again awash, the heating chamber is once more used, and the water taken up by the soda solution will be returned to the condenser. Cocks and valves are fitted as found necessary.

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